TRANSACTIONS

American Society of Heating and Ventilating Engineers

VOLUME 38

THIRTY-EIGHTH ANNUAL MEETING
CLEVELAND, OHIO, JANUARY 25-29, 1932

THIRTY-EIGHTH SEMI-ANNUAL MEETING
MILWAUKEE, WIS., JUNE 27-29, 1932

PUBLISHED BY THE SOCIETY AT THE OFFICE OF THE SECRETARY
51 MADISON AVENUE
NEW YORK, N. Y., U. S. A.

PRINTED IN U.S.A.

CONTENTS

Officers and Council—1932

American Society *of* Heating *and* Ventilating Engineers

President	F. B. Rowley
First Vice-President	W. T. Jones
Second Vice-President	C. V. Haynes
Treasurer	F. D. Mensing
Secretary	A. V. Hutchinson
Technical Secretary	Paul D. Close

Council

F. B. Rowley, *Chairman* W. T. Jones, *Vice-Chairman*

One Year	*Two Years*	*Three Years*
D. S. Boyden	E. K. Campbell	F. E. Giesecke
R. H. Carpenter	E. O. Eastwood	G. L. Larson
W. H. Carrier	Roswell Farnham	J. F. McIntire
J. D. Cassell	E. H. Gurney	W. E. Stark
John Howatt		

Committees of the Council

Executive: W. T. Jones, *Chairman;* W. H. Carrier, E. H. Gurney.

Finance: John Howatt, *Chairman;* R. H. Carpenter, W. T. Jones.

Membership: Roswell Farnham, *Chairman;* J. D. Cassell, W. E. Stark.

Publication: F. D. Mensing, *Chairman;* E. O. Eastwood, F. E. Giesecke.

Advisory Council

W. H. Carrier, *Chairman;* Homer Addams, F. Paul Anderson, R. P. Bolton, S. E. Dibble, W. H. Driscoll, H. P. Gant, John F. Hale, L. A. Harding, H. M. Hart, E. Vernon Hill, J. D. Hoffman, S. A. Jellett, D. D. Kimball, S. R. Lewis, Thornton Lewis, J. I. Lyle, J. R. McColl, D. M. Quay, C. L. Riley, F. R. Still and A. C. Willard.

Committees—1932

COMMITTEE ON RESEARCH

G. L. Larson, *Chairman*
Perry West, *Vice-Chairman*

F. C. Houghten, *Director* O. P. Hood, *Ex-Officio Member*
A. C. Willard, *Technical Adviser*

One Year	*Two Years*	*Three Years*
C. V. Haynes	C. A. Booth	D. E. French
W. T. Jones	F. B. Howell	F. E. Giesecke
J. F. McIntire	Walter Klie	A. P. Kratz
F. N. Speller	F. C. McIntosh	G. L. Larson
Perry West	J. H. Walker	L. A. Harding

COMMITTEE ON RESEARCH (*continued*)

Executive Committee

G. L. Larson, *Chairman*

J. F. McIntire W. T. Jones

Finance Committee

S. R. Lewis, *Chairman*

W. H. Carrier C. V. Haynes F. B. Howell J. H. Walker

Technical Advisory Committees

Air Conditions and Their Relation to Living Comfort: C. P. Yaglou, *Chairman;* I. C. Baker, W. L. Fleisher, D. E. French, Dr. E. V. Hill and Dr. R. R. Sayers.

Air Flow Through Registers and Grilles: John Howatt, *Chairman;* J. J. Aeberly, C. A. Booth, L. E. Davies, D. E. French and J. J. Haines.

Atmospheric Dust and Air Cleaning Devices (Including Dust and Smoke): H. C. Murphy, *Chairman;* J. J. Aeberly, J. J. Bloomfield, Albert Buenger, Philip Drinker, Dr. E. V. Hill, H. B. Meller, Games Slayter, Perry West and Dr. S. W. Wynne.

Correlating Thermal Research: R. M. Conner, *Chairman;* D. S. Boyden, P. D. Close, J. C. Fitts, W. T. Jones, H. T. Richardson and Perry West.

Direct and Indirect Radiation with Gravity Air Circulation: H. F. Hutzel, *Chairman;* R. M. Conner, R. E. Daly, R. V. Frost, J. H. Holton, A. P. Kratz, J. F. McIntire, W. T. Miller, R. N. Trane and O. G. Wendel.

Gas Heating Equipment: W. E. Stark, *Chairman;* R. M. Conner, Robert Harper, F. B. Howell, E. A. Jones, Thomson King, Henry Loebell, J. F. McIntire, L. S. Ourusoff and H. L. Whitelaw.

Heat Transmission (Heat received by and emitted by buildings, in relation to living comfort): L. A. Harding, *Chairman;* A. B. Algren, R. E. Backstrom, A. H. Barker, A. E. Stacey, Jr., and J. H. Walker.

Infiltration in Buildings: J. G. Shodron, *Chairman;* J. E. Emswiler, F. E. Giesecke, D. W. Nelson, W. C. Randall, Ernest Szekely, J. H. Walker and M. S. Wunderlich.

Oil Burning Devices: H. F. Tapp, *Chairman;* P. E. Fansler, R. V. Frost, F. B. Howell, R. C. Morgan and L. E. Seeley.

Pipe and Tubing Carrying Low Pressure Steam or Hot Water: S. R. Lewis, *Chairman;* J. C. Fitts, F. E. Giesecke, H. M. Hart, C. A. Hill, A. P. Kratz, R. R. Seeber and W. K. Simpson.

Refrigeration in Relation to Air Treatment: A. P. Kratz, *Chairman;* E. A. Brandt, Geo. B. Bright, S. R. Lewis, H. J. Macintire, E. D. Milener, F. G. Sedgwick, A. R. Stevenson, Jr., J. H. Walker, R. W. Waterfill, A. C. Willard, A. W. Williams and H. M. Williams.

Re-Study of Comfort Chart and Comfort Line: C. P. Yaglou, *Chairman;* W. H. Carrier, E. V. Hill, F. C. Houghten and J. H. Walker.

Sound in Relation to Heating and Ventilation: Warren Ewald, *Chairman;* Carl Ashley, C. A. Booth, V. O. Knudsen, R. F. Norris, J. P. Reis and G. T. Stanton.

Ventilation of Garages and Bus Terminals: E. K. Campbell, *Chairman;* T. M. Dugan, E. C. Evans, L. A. Harding, F. H. Hecht, V. W. Hunter, F. C. McIntosh, H. Lee Moore and A. H. Sluss.

Special Committees

Committee on Chapter Relations: E K. Campbell, *Chairman;* Roswell Farnham and J. F. McIntire.

Committee on Code for Testing and Rating Concealed Gravity Type Radiation: R. N. Trane, *Chairman;* E. H. Beling, J. H. Holton, Hugo Hutzel, A. P. Kratz, J. F. McIntire, M. G. Steele and O. G. Wendel.

Committee on Code for Testing and Rating Condensation and Vacuum Pumps: John Howatt, *Chairman;* W. H. Driscoll, R. E. Gilmore and L. A. Harding.

Committee on Meetings Program: A. P. Kratz, *Chairman;* L. A. Harding and C. V. Haynes.

Committee on Revision of Constitution and By-Laws: W. T. Jones, *Chairman;* and Thornton Lewis.

Committee on Ventilation Standards: W. H. Driscoll, *Chairman;* J. J. Aeberly, F. Paul Anderson, L. A. Harding, D. D. Kimball, J. R. McColl, C. L. Riley, W. A. Rowe, Perry West and A. C. Willard.

Guide Publication Committee: D. S. Boyden, *Chairman;* E. O. Eastwood, W. L. Fleisher, S. R. Lewis and Perry West.

Nominating Committee:

Cleveland Chapter—C. F. Eveleth
Illinois Chapter—H. M. Hart
Kansas City Chapter—Benjamin Natkin
Massachusetts Chapter—C. P. Yaglou
Michigan Chapter—G. D. Winans
Western Michigan Chapter—S. H. Downs
Minnesota Chapter—C. E. Lewis
New York Chapter—E. J. Ritchie
Western New York Chapter—Joseph Davis
Ontario Chapter—W. G. Sheppard
Philadelphia Chapter—M. F. Blankin
Pittsburgh Chapter—F. C. McIntosh
St. Louis Chapter—C. A. Pickett
Southern California Chapter—L. H. Polderman
Wisconsin Chapter—J. G. Shodron

Officers of Local Chapters—1932

Cincinnati

Headquarters, Cincinnati
Meets: Second Tuesday in Month

President, K. A. Wright
Vice-President, H. N. Kitchell
Secretary-Treasurer, G. B. Houliston
Board of Governors: G. B. Houliston, C. J. Kiefer, H. N. Kitchell, E. B. Royer, K. A. Wright

Cleveland

Headquarters, Cleveland
Meets: Second Friday in Month

President, R. G. Davis
Vice-President, L. H. Pogalies
Secretary, F. A. Kitchen
Treasurer, H. M. Nobis
Board of Governors: W. E. Stark, Vincent Eaton, C. J. Deex

Illinois

Headquarters, Chicago
Meets: Second Monday in Month

President, J. J. Aeberly
Vice-President, J. H. O'Brien
Secretary, C. W. DeLand
Treasurer, C. W. Johnson
Board of Governors: H. P. Reid, Eugene Mathis, A. G. Sutcliffe

Kansas City

Headquarters, Kansas City, Mo.
Meets: Second Monday in Month

President, W. A. Russell
Vice-President, David Caleb
Secretary, D. D. Zink
Treasurer, Emil Haas
Board of Governors: J. M. Arthur, H. D. Betz, C. A. Weiss

Massachusetts

Headquarters, Boston
Meets: Second Monday in Month

President, F. R. Ellis
Secretary-Treasurer, Leslie Clough
Board of Governors: Leslie Clough, R. F. Gifford, F. R. Ellis, C. R. Swaney, E. W. Berchtold, L. D. Cushman

Michigan

Headquarters, Detroit
Meets: First Monday after the 10th of the Month

President, L. L. McConachie
Vice-President, H. E. Paetz
Secretary, G. D. Winans
Treasurer, E. H. Clark
Board of Governors: R. K. Milward, J. W. Snyder, A. C. Wallich

Western Michigan

Headquarters, Grand Rapids
Meets: Second Monday in Month

President, O. D. Marshall
Vice-President, D. L. Taze
Secretary, C. H. Alexander
Treasurer, K. L. Ziesse
Board of Governors: W. W. Bradfield, W. J. Carroll, E. L. Jaynes

Minnesota

Headquarters, Minneapolis
Meets: Second Monday in Month

President, W. F. Uhl
Vice-President, A. B. Algren
Secretary-Treasurer, C. E. Lewis
Board of Governors: H. E. Gerrish, C. E. Gausman

New York

Headquarters, New York
Meets: Third Monday in Month

President, L. T. M. Ralston
Vice-President, H. L. Alt
Secretary, T. W. Reynolds
Treasurer, W. M. Heebner
Board of Governors: Russell Donnelly, L. K. Berman, Rudolf Kuhlmann

New York University

Student Chapter
Headquarters, New York University

President, William Rozett, Jr.
Vice-President, Herbert Maiman
Secretary, H. F. Austin, Jr.
Treasurer, J. C. Bruckmann

Western New York

Headquarters, Buffalo
Meets: Second Monday in Month

President, M. C. Beman
First Vice-President, D. J. Mahoney
Second Vice-President, J. J. Yager
Secretary, W. E. Voisinet
Treasurer, W. F. Johnson
Board of Governors: R. T. Coe, L. A. Harding, C. W. Farrar, H. F. Hutzel, W. G. Fraser, Roswell Farnham, M. S. Jackson, C. A. Evans, O. K. Dyer, F. H. Burke, Joseph Davis

Ontario

Headquarters, Toronto, Ont.
Meets: First Monday Every Other Month

President, J. S. Wood
Vice-President, W. P. Boddington
Secretary-Treasurer, H. R. Roth
Board of Governors: M. W. Shears, J. S. Paterson, W. R. Blackhall, H. S. Moore

Pacific Northwest

Headquarters, Seattle, Wash.
Meets: Second Thursday in Month

President, C. F. Twist
Vice-President, P. M. O'Connell
Secretary, S. D. Peterson
Treasurer, W. W. Cox
Board of Governors: W. E. Beggs, E. O. Eastwood, E. L. Weber

Philadelphia

Headquarters, Philadelphia
Meets: Second Thursday in Month

President, L. C. Davidson
Vice-President, M. F. Blankin
Secretary, W. P. Culbert
Treasurer, J. H. Hucker
Board of Governors: E. N. Sanbern, H. F. Rettew, W. R. Eichberg

Pittsburgh

Headquarters, Pittsburgh
Meets: First Monday in Month

President, R. B. Stanger
Vice-President, F. C. McIntosh
Secretary, J. L. Blackshaw
Treasurer, H. B. Orr
Board of Governors: F. C. Houghten, G. S. McElroy, H. L. Moore

St. Louis

Headquarters, St. Louis
Meets: First Wednesday in Month

President, J. M. Foster
First Vice-President, C. A. Pickett
Second Vice-President, P. W. Sodemann
Secretary, C. R. Davis
Treasurer, A. L. Walters
Board of Governors: R. M. Rosebrough, E. A. White, G. W. F. Myers, L. W. Moon

Southern California

Headquarters, Los Angeles
Meets: Second Tuesday in Month

President, F. R. Winch
Vice-President, L. H. Polderman
Secretary, E. H. Kendall
Treasurer, Leo Hungerford
Board of Governors: M. M. Lawler, W. H. C. Ness, O. W. Ott

Wisconsin

Headquarters, Milwaukee
Meets: Third Monday in Month

President, E. A. Jones
First Vice-President, V. A. Berghoefer
Second Vice-President, J. S. Jung
Secretary, H. F. Haupt
Assistant Secretary, A. F. Bowers
Treasurer, M. E. Erickson
Board of Governors: G. L. Larson, F. G. Weimer, J. G. Shodron

TRANSACTIONS
of
AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

No. 912

THIRTY-EIGHTH ANNUAL MEETING, 1932

A BUSINESS session at which it was revealed that the Society came through 1931 in excellent financial condition, two joint sessions with the *American Society of Refrigerating Engineers,* a *code* session at which the reports of two important committees were presented, and a **registration of** 725 members and guests were the high lights of the 38th Annual Meeting of the Society, held in Cleveland, Ohio, January 26-29, 1932.

Of the fifteen scientific and technical papers presented nine resulted from research conducted in the A.S.H.V.E. Laboratory at Pittsburgh, and at institutions with which the Society maintains co-operative agreements.

Pres. W. H. Carrier called the meeting to order in Hotel Statler and W. E. Stark, president of the Cleveland Chapter, made the address of welcome.

President Carrier then read the report of the Council and in this connection stated that Secretary Hutchinson had taken the initiative in endeavoring to render aid to members of the Society who were out of employment, and in some cases, in distress.

Report of the Council

Meetings of the Council were held in January, April, June and November, 1931, and January, 1932, and all business matters prescribed in the Constitution and By-Laws were acted upon.

The organization meeting was held in Pittsburgh, Pa., January 29, 1931, with Pres. W. H. Carrier presiding and standing and special committee appointments were announced and depositories for Society funds were selected.

On April 13, the Council met in Chicago, Ill., and formally adopted the Budget for 1931 and approved the program for the Semi-Annual Meeting at Swampscott.

The idea of quarterly publication of TRANSACTIONS was considered and referred to the Publication Committee for study and recommendation.

At the Semi-Annual Meeting at Swampscott two Council meetings were held. The

invitation of the Wisconsin Chapter to have the Semi-Annual Meeting 1932 was accepted during the week of June 27. The matter of amending the Society's Constitution and By-Laws was discussed in detail and it was also recommended that the rules for procedure prepared by a special committee be referred to the Committee on Revision of the Constitution with a suggestion that the rules be made an appendix to the Constitution and By-Laws.

Nominees for membership on the Committee on Research were made. A Committee on Award for the F. Paul Anderson Medal was announced and members of the Society were invited to submit the names of nominees for the award.

The idea of a co-operative program to obtain speakers for several chapters was proposed by the Chapter Relations Committee and approved.

At a special meeting of the Council, the budget of the Committee on Research was approved, and Perry West was designated to serve as a member of the Committee on Research in place of Prof. A. C. Willard.

The idea of continuing to publish the JOURNAL as a section of *Heating, Piping and Air Conditioning* was thoroughly discussed and the matter was referred to the Committee who originally negotiated the contract, with power to make the best possible terms. An agreement for a five-year period was prepared and signed.

In Buffalo, November 9, the Council granted the petition of the Western Michigan members for a Chapter at Grand Rapids; received the report of the Anderson Medal Committee, and approved the program for the Annual Meeting 1932. Reports from various special committees were received including the proposed Code for Testing and Rating Unit Ventilators; the revised contract for publishing the JOURNAL; the quarterly publication of TRANSACTIONS and the publication of THE GUIDE.

At all meetings, the matter of relief for members affected by economic conditions was a matter of serious discussion and every encouragement was given to men whose dues were in arrears, to continue their membership.

Formal action was taken in granting life membership to 13 men, reinstatement of 13 men, and the acceptance of resignations of 74. Action was taken with regret in the case of 21 who failed to affiliate and 107 who were in arrears for dues.

Report of the Secretary

The Society has passed through the trying year 1931 with a record of many things accomplished, a slight increase in membership during the calendar year and with a fair financial outlook for the beginning of the new year. The progress made during the fiscal year in handling a greatly increased volume of work without enlarging the staff, will give an indication of the splendid co-operation of my assistants who have so ably handled their assignments and have spent many extra hours on the job without additional compensation.

The various committees that have functioned during this year have handled their work with dispatch and the results of the progress of their deliberations will be reported at this meeting.

Some of the important actions taken during the year were the approval by the Society of two new Codes and the revision of one previously adopted:

1. Code for Testing Steam Boilers Burning Oil Fuel.
2. The Code for Testing and Rating Concealed Gravity Type Radiation.
3. Revision of Standard Code for Testing Centrifugal Blowers and Disc Fans.

Since the last Annual Meeting a special committee has compiled the Code for Testing and Rating Unit Ventilators to be presented here for consideration and a set of ventilation standards has been prepared by a special committee to meet the demands of modern requirements of various types of buildings. Discussion of these two reports is to be scheduled at this meeting.

In the publication work the items of major importance were—

1. Production and distribution of THE GUIDE 1932.
2. Compilation of the TRANSACTIONS 1930 (ready for mailing).

3. Printing of the TRANSACTIONS INDEX for the years 1895 to 1930.
4. Reprinting and distribution of four standard codes previously adopted by the Society—
   a. Standard Code for Testing Solid Fuel Boilers.
   b. Performance Test Code for Boilers Burning Solid Fuel.
   c. Standard Code for Testing and Rating Steam Unit Heaters.
   d. Standard Code for Testing Heat Transmission of Building Materials.
5. Production of the booklet describing the research work of the Society.

An effort that has developed to one of major importance resulted from the employment survey made among 2,000 manufacturing, engineering and contracting firms with the object of keeping trained heating and ventilating engineers employed in their own specialized field. The plan developed was to obtain leads and refer them to members who were seeking new connections. Chapter representatives were assigned by the various local organizations to assist in locating the men who needed work so that proper leads and interviews might be arranged. Both direct-by-mail and direct advertising through the trade press is being used to assist the members. Some progress has been made in placing men in various parts of the country. There is still a long list of those who need assistance, particularly men who have specialized in design, layout and estimating.

The Council granted a Charter for the Western Michigan Chapter at Grand Rapids and this group formed a permanent organization after a preliminary meeting in October attended by the President, Secretary and Messrs. McColl, Rowe and McIntire of Detroit.

Each Chapter of the Society was visited by the President and Secretary during the year in the 16 cities where local organizations are functioning.

The present status of membership shows a total of 2,256: 2 honorary members, 1,539 members, 490 associates, 203 juniors and 22 students, an increase of 27 during the current year.

Report of the Treasurer

The report of the treasurer was submitted verbally by F. D. Mensing who stated that the financial returns for 1931 were satisfactory considering the circumstances. The returns for dues were slightly in excess of the previous year, whereas the returns for new members were slightly under the previous year. Mr. Mensing stated that the delinquent members were written to individually and the response was very satisfactory.

Report of the Finance Committee

The next report was that of F. C. McIntosh, chairman of the Finance Committee, who stated that there was a good profit from THE GUIDE and a net loss on general Society operations of only $138, which he considered exceptionally good for the year 1931. Although the income from dues and miscellaneous sources was far under the budget and under what could be expected in normal times, Mr. McIntosh said, the Society was able to effect considerable savings in general expenses through the persistent effort of Secretary Hutchinson.

On motion of J. J. Aeberly, which was seconded by John Howatt, it was voted that the reports of the Treasurer and the chairman of the Finance Committee be accepted.

Report of Certified Public Accountant

Jaunary 8, 1932.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS,
51 MADISON AVENUE,
NEW YORK CITY.

Gentlemen:

Pursuant to your request I made an examination of the books of account and records of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York City, for the year ended December 31, 1931, and submit herewith my report.

The work covered a verification of the Assets and Liabilities as of the date previously stated and a review of the operating accounts for the calendar year 1931. For the aforementioned period, the recorded cash receipts were traced into the depositories; the cancelled bank vouchers were compared with the cash disbursement records; and membership dues were accounted for.

Submitted herewith is a Balance Sheet showing the financial condition of the Society on December 31, 1931 and your attention is directed to the following comments thereon:

CASH

Cash on Deposit was verified by direct communication with the banks listed below and by reconcilement of the amounts reported to me with the balances shown by the books of the Society.

BANKS	AMOUNTS
Chase National Bank (Special)	$ 39.63
Chase National Bank (Regular)	140.90
Bankers Trust Company	1,842.86
Bowery Savings Bank (Book No. 57,001)	642.63
Bank of United States in Liquidation	500.00
Total	$3,166.02

The petty cash on hand was counted and found correct.

MARKETABLE SECURITIES

There is attached hereto a schedule of negotiable bonds which were verified by direct communication with the Bankers Trust Company where same are deposited for safekeeping. No adjustment has been made of the $12,293.17 shrinkage in the market value of these securities. These have been included on the attached Balance Sheet at cost.

ACCOUNTS RECEIVABLE

Unpaid membership dues were determined by trial balance of the individual ledger cards and comparison of same with the Dues Register. A summary of them as to years charged is shown below:

1931 Unpaid Dues	$12,743.75
1930 Unpaid Dues	4,154.09
1929 Unpaid Dues	760.00
Total	$17,657.84

My verification of the dues disclosed that during the year 1931 there had been prepaid to the Society dues amounting to $206.75 which I have shown on the attached Balance Sheet as deferred income.

The Reserve found on the books to cover probable losses which may result from prior years' dues was found ample. In addition, however, I have provided the sum of $9,557.81 which is equivalent to seventy-five per cent (75%) of the unpaid 1931 dues. Adequate Reserves have also been provided to cover losses which may be incurred from the realization of all other accounts receivable.

BALANCE SHEET

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

December 31, 1931

ASSETS

SOCIETY				
CASH				
On Deposit		$ 3,166.02		
On Hand		100.00	$ 3,266.02	
INVESTMENTS (AT COST)				
Securities (Market Value $10,540.00)		12,919.49		
ADD: Accrued Interest		78.75	12,998.24	
ACCOUNTS RECEIVABLE				
Membership Dues	$17,657.84			
LESS: Reserve for Doubtful	14,537.57	3,120.27		
Advertisements	39,083.30			
LESS: Reserve for Doubtful	6,000.00	33,083.30		
Other	3,714.62			
LESS: Reserve for Doubtful	1,000.00	2,714.62	38,918.19	
INVENTORIES				
Transactions 1924-1929		1,870.87		
Transactions 1930 in Process		706.32		
Emblems and Certificate Frames		113.20		
Postage		644.20	3,334.59	
PERMANENT				
Library		300.00		
Furniture and Fixtures	5,165.61			
LESS: Reserve for Depreciation	2,216.47	2,949.14	3,249.14	
DEFERRED CHARGES				
Meetings 1932			149.58	$61,915.76
SPECIFIC FUNDS				
ENDOWMENT FUND				
Securities at Cost (Market Value $10,237.50)	19,753.68			
ADD: Accrued Interest	250.83	20,004.51		
Cash on Hand for Deposit		1,025.00	21,029.51	
F. PAUL ANDERSON AWARD FUND				
Cash on Deposit		337.90		
Cash on Hand for Deposit		685.53	1,023.43	
RESEARCH LABORATORIES FUND				
Cash on Deposit—Research	8.25			
Cash on Deposit—Research Endowment Fund	335.53			
Cash on Hand for Deposit Endowment Fund	1,724.51	2,068.29		
Securities at Cost (Market Value $2,610.00)	3,007.50			
ADD: Accrued Interest	45.00	3,052.50	5,120.79	27,173.73
				$89,089.49

LIABILITIES AND CAPITAL

SOCIETY			
ACCOUNTS PAYABLE		$10,984.33	
DUE RESEARCH LABORATORY			
Dues		6,491.19	
ACCRUED ACCOUNTS			
Bonus to Staff....................		3,008.62	
RESERVE FOR TRANSACTIONS			
1930	$ 3,000.00		
1931	4,000.00	7,000.00	
DEFERRED INCOME			
Prepaid Dues		206.75	
TOTAL LIABILITIES		27,690.89	
GENERAL FUND			
Society		34,224.87	$61,915.76
SPECIFIC FUNDS			
Endowment		21,029.51	
F. Paul Anderson Award..........		1,023.43	
Research Laboratory	4,510.75		
Research Endowment Fund........	610.04	5,120.79	27,173.73
			$89,089.49

Note "A": The General Fund of the Society is subject to future adjustment pending action to be taken by Council on the assignment of THE GUIDE Profit for the Calendar Year 1931.

Note "B": This Balance Sheet is subject to the comments contained in the letter attached to and forming a part of this report.

INVENTORIES

There is scheduled below the inventory of TRANSACTIONS taken on December 31, 1931.

YEAR	VOLUME	NUMBER	PRICE	AMOUNT
1924	30	273	138¼	$ 377.42
1925	31	168	96⅙	161.56
1926	32	106	126⅘	134.41
1927	33	154	132⅗	204.21
1928	34	224	143	320.32
1929	35	313	215	672.95
Total..				$1,870.87

TRANSACTIONS covering the year 1930, volume 36, and the year 1931, Volume 37, were not published up to the close of the current year, therefore reserves of $3,000.00 and $4,000.00 respectively have been provided to cover the future cost thereof.

All other inventories were verified either by actual count or analysis of the records.

ACCOUNTS PAYABLE

A list was compiled of all invoices remaining unpaid on December 31, 1931, dating prior to January 1, 1932, for the purpose of determining all Accounts Payable, and a liability therefor was set up in the sum of $10,984.33. This sum includes charges from Horn-Shafer Co. of $9,851.29 for costs covering the printing of THE GUIDE and Year Book.

DUE RESEARCH LABORATORY

Of the dues charged to Seniors and Associates, 40% has been reserved for the Research Laboratory in accordance with Section 5, Article 3, of the By-Laws. The sum payable to the Research Laboratory as and when the Dues Receivable will have been realized in cash is $6,491.19.

ACCRUED ACCOUNTS

A bonus of $3,008.62 to the staff of the Society has been included as a liability on the subjoined Balance Sheet. The computation of this bonus has been made in accordance with resolution adopted by Council.

GENERAL FUND

An analysis of the General Fund of the Society showing the changes made therein during the year under audit is as follows:

General Fund, January 1, 1931—per former report		$32,188.72
Addition		
Profit from Guide for the Calendar Year 1931 from Guide Statement of Income and Expenses		3,154.75
		$35,343.47
Deductions		
Loss from Society activities for the Calendar Year 1931 from Society Statement of Income and Expenses	*$1,093.60*	
To charge General Fund with refund of 1930 dues to life member	*25.00*	1,118.60
General Fund December 31, 1931—per balance sheet		$34,224.87

MEMBERSHIP

A comparison of the membership in force as of the close of business December 31, 1930 and 1931 respectively, follows:

CLASSIFICATION	1931	1930	Increases *Decreases*
Members	1,539	1,485	54
Associates	490	480	10
Juniors	203	246	*43*
Students	22	16	6
Honorary	2	2	0
Totals	2,256	2,229	27

Respectfully submitted,

FRANK G. TUSA,
Certified Public Accountant.

BUDGET COMPARISON—GUIDE

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK CITY

For the Year Ended December 31, 1931

	ACTUAL	BUDGET PROVISION	Increases *Decreases*
INCOME			
Advertising	$32,760.76	$38,000.00	*$5,239.24*
Sales	11,778.28	14,500.00	*2,721.72*
TOTALS	$44,539.04	$52,500.00	*$7,960.96*
EXPENSES			
Printing and Binding 1932 Issue	$11,335.74	$13,500.00	*$2,164.26*
Mailing 1932 Issue	300.00	1,500.00	*1,200.00*
Paper Purchases	2,482.07	2,800.00	*317.93*
Engraving and Art Work	469.03	600.00	*130.97*
Sales Promotion	2,702.73	2,000.00	702.73
Postage and Mailing 1931 Issue	2,033.83	1,600.00	433.83
Salaries (Except Secretary)	7,469.03	7,300.00	169.03
25% Apportionable Expenses	4,583.24	4,925.00	*341.76*
TOTALS	$31,375.67	$34,225.00	*$2,849.33*

BUDGET COMPARISON—SOCIETY ACTIVITIES

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK CITY

For the Year Ended December 31, 1931

	ACTUAL 1931	BUDGET PROVISION	Increases *Decreases*
INCOME			
Current Dues—Less Reserve	$25,798.73	$28,000.00	*$2,201.27*
Initiation Fees	2,213.00	4,000.00	*1,787.00*
Sales—Emblems and Certificate Frames	219.50	350.00	*130.50*
Sales—Codes	5.00	100.00	*95.00*
Sales—Journal and Non-Member Subscriptions	73.10	100.00	*26.90*
Sales—Reprints and Books	868.36	1,000.00	*131.64*
Sales—Transactions	642.25	800.00	*157.75*
Editorial Contract—Less Subscriptions	12,518.63	13,500.00	*981.37*
Interest Earned—General Funds	620.00	620.00	—0—
Interest Earned—Current Funds	186.43	280.00	*93.57*
TOTALS	$43,145.00	$48,750.00	*$5,605.00*
EXPENSES			
Salaries (Exclusive of Secretary)	$12,503.15	$12,000.00	$ 503.15
Postage	2,038.96	1,800.00	238.96
General Printing	895.03	1,000.00	*104.97*
Year Book	949.98	900.00	49.98
Transactions—1931	5,373.25	3,500.00	1,873.25
Reprints and Books	603.68	800.00	*196.32*
Emblems and Certificate Frames	205.82	300.00	*94.18*
Meetings—Annual and Semi-Annual	2,807.38	3,000.00	*192.62*
Meetings—Council and Special Committees	607.79	1,000.00	*392.21*
Traveling—President	1,000.00	1,000.00	—0—
Dues	60.00	60.00	—0—
Codes	660.68	1,000.00	*339.32*
Publicity	902.72	1,000.00	*97.28*
President's Emergency Fund	445.80	500.00	*54.20*
75% of Apportionable Expenses	13,749.70	14,775.00	*1,025.30*
	42,803.94	42,635.00	168.94
COMMITTEE ON RESEARCH EXPENSES	955.59	—0—	955.59
UNBUDGETED EXPENSES			
Transactions Index	278.57	—0—	278.57
Membership Campaign Expenses	111.50	—0—	111.50
Engraving Certificates	89.00	—0—	89.00
	$44,238.60	$42,635.00	$1,603.60
	$ 1,093.60	$ 6,115.00	*$7,208.60*

Report of Publication Committee

The report of the Publication Committee was submitted by W. A. Rowe, chairman, who stated that about 25 papers had been scheduled for Society meetings, approximately equally divided between the Summer Meeting 1931 and the Annual Meeting 1932. Nine papers had been rejected or had been withdrawn by the authors. Mr. Rowe enumerated some of the publications which had been sent out during the year and expressed his appreciation for the high type of papers that had been submitted during his chairmanship.

Report of the Committee on Research

The Report of the Committee on Research was submitted by C. V. Haynes, chairman, and F. C. Houghten gave the report of the Director.

The Committee on Research of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS gave consideration to 12 research projects during the year. In connection with each of these ventures, a sub-committee of the Committee on Research,

known as a Technical Advisory Committee, was appointed for the purpose of studying the needs of the problem, and to cooperate with the Director of the Laboratory in plans for research to be carried on in the Laboratory at Pittsburgh or in the cooperating institutions. Each of these Technical Advisory Committees was made up of authorities on the particular subject. During the past year this organization brought to the Society's research activity the combined experience and ability of 78 experts.

The Committee on Research was very fortunate in securing the services of Prof. A. C. Willard as Technical Adviser to the Committee on Research. This addition to the organization is of great value, as it brings the broad experience and ability of this noted engineer and educator to aid in the work. Other slight changes in organization were made in order to better correlate the research activities with the other activities of the Society.

Five meetings of the Committee on Research were held during the year: two during the Annual Meeting in Pittsburgh, one in Chicago in the spring, one at Swampscott during the Summer Meeting of the Society, and the final meeting at Buffalo in November. At these meetings the welfare of the Laboratory was discussed including its finances, the development of a research program, and the approval of cooperative agreements for research.

The funds for the Laboratory work were definitely allocated in the budget and as in the past were derived from dues of Society members, interest on reserve funds, THE GUIDE, and contributions from the industry, which were most generous considering present financial conditions. This made a total commensurate with previous years. New contracts and greater contributions to institutions cooperating with the Laboratory made it necessary to slightly revise the budgeted expenditures at Pittsburgh.

Cooperative Research

Cooperative research programs have been arranged for by contract with the following institutions where the projects listed are being investigated:

University of Minnesota: Heat transmission through built-up walls, determination of surface coefficients, conductivity of insulating materials, and the thermal properties of different species of wood. The latter study is carried on through the cooperation of the *National Lumber Manufacturers Association* with the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the University of Minnesota.

University of Illinois: The study of the performance characteristics of radiators.

Harvard School of Public Health: The ionization of air in its relation to health.

University of Wisconsin: The aeration of buildings due to infiltration and other air exchanges.

Yale University: Development of standard test methods for oil burning devices in steam and hot water heating boilers.

Carnegie Institute of Technology: Study of steam, condensate and air flow in low pressure steam heating systems.

Agricultural and Mechanical College of Texas: Frictional resistance to the flow of water in hot water heating systems.

Armour Institute of Technology: Measurement of air flow through registers and grilles, including air delivery from unit ventilators.

Research Subjects Considered

Of the 12 research subjects given consideration by the Committee on Research and its Technical Advisory Committees, 7 were actively investigated at the Laboratory in Pittsburgh or in one of the cooperating institutions. From these studies, 16 papers were prepared for publication and presentation at either the 1931 Summer or the 1932 Annual Meeting of the Society.

A paper, Heat and Moisture Dissipation from Children in Relation to School Ventilation, by F. C. Houghten, W. W. Teague, and W. E. Miller, was presented at the Annual Meeting of the *American Public Health Association* in Montreal in September, 1931.

The following 12 subjects were given consideration during the past year by the Committee on Research and the Technical Advisory Committees.

1. HEAT TRANSMISSION—(Heat Received and Emitted by Buildings in Relation to Living Comfort).—*Technical Advisory Committee:* L. A. Harding, *Chairman;* R. E. Backstrom, F. B. Rowley, A. E. Stacey, J. H. Walker.

During the past year studies have been made at the University of Minnnesota, (*a, b, c* and *d*) in cooperation with the Research Laboratory and at the Laboratory in Pittsburgh, (*e, f* and *g*).

a. A STUDY OF THE THERMAL CONDUCTIVITIES FOR VARIOUS TYPES OF BUILT-UP WALL CONSTRUCTION BY THE HOT-BOX METHOD.
b. STUDY ON SURFACE COEFFICIENTS WITH WIND NOT PARALLEL TO THE WALL.
c. THERMAL CONDUCTIVITY OF INSULATING MATERIALS.
d. CONDUCTIVITY OF VARIOUS SPECIES OF LUMBER.
e. VARIATION IN CONDUCTIVITY OF CONCRETE.
f. EFFECT OF SOLAR RADIATION AND HEAT CAPACITY OF A STRUCTURE ON HEAT FLOW INTO A BUILDING.
g. WIND VELOCITY GRADIENTS NEAR A WALL.
h. THE COOLING DEMAND IN AN OFFICE BUILDING DURING THE SUMMER. (Work carried on by J. H. Walker in cooperation with the Research Laboratory.)

2. PIPE SIZES—Pipe and Tubing (Sizes) Carrying Low Pressure Steam and Hot Water.—*Technical Advisory Committee:* S. R. Lewis, Chairman; J. C. Fitts, F. E. Giesecke, H. M. Hart, C. A. Hill, A. P. Kratz, W. K. Simpson.

During the year the Laboratory in Pittsburgh made an analysis of THE GUIDE tables. As a result of this analysis, the Technical Advisory Committee at a meeting during the semi-annual meeting of the Society agreed that the following information was still desirable before making a final revision of THE GUIDE tables:

First, air and condensation loads on the return side of gravity return and vacuum return steam heating systems both during and after the heating-up period; *Second,* determination of the capacity of 1½-in. and 2-in. dry return mains, risers and wet return mains in gravity and vacuum pump systems.

During the past year, the Laboratory in cooperation with the *Copper Tubing Manufacturers Association* has made a study of the performance of the copper tubing in steam and hot water heating systems. The studies relating to steam heating systems were carried on in Pittsburgh, and those relating to hot water systems were carried on under the direction of Professor Giesecke at the Agricultural and Mechanical College of Texas in cooperation with the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

The study of performance characteristics of copper tubing in steam heating systems included: *First,* a study of the heat emission from the surface of bare and insulated copper tubing and iron pipe; *Second,* a study of the relative capacity of copper tubing for carrying steam and condensate in vertical and horizontal straight runs of pipe and in radiator supply branches. These studies on copper tubing have been completed.

The study of frictional resistance to flow of water in hot water heating systems was continued by Professor Giesecke of the Agricultural and Mechanical College of Texas in cooperation with the Research Laboratory. The present investigation includes a study of the frictional resistance in orifices and standard iron tees and frictional resistance in copper tubes and fittings.

3. AIR CLEANING—Atmospheric Dust and Air Cleaning Devices (Including Dust and Smoke).—*Technical Advisory Committee:* H. C. Murphy, *Chairman;* Albert Buenger, Philip Drinker, Dr. E. V. Hill, H. B. Meller, Dr. S. W. Wynne.

Early during the present year a study was carried on by Dean Langsdorf of Washington University, St. Louis, in cooperation with the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. The object of this study was to develop a more sensitive and more satisfactory means of determining the dustiness of air. The method was rather completely developed by Dean Langsdorf, but the instrument is not yet completely calibrated and tested out for practical use.

4. AIR FLOW—Air Flow Through Registers and Grilles.—*Technical Advisory Committee:* John Howatt, *Chairman;* J. J. Aeberly, C. A. Booth, L. E. Davies, D. E. French, J. J. Haines.

The study of methods of measuring air flow through registers and grilles is being carried on by Professor Davies at Armour Institute of Technology in cooperation with the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *Ventilating Contractors Employers Association* of Chicago. The development of the work to date has covered four phases [1, 2].

The investigation will be continued at Armour Institute of Technology and will include the following three phases:

1. Ornamental and other special types of grilles. Investigations last year showed that on supply grilles when the designs were characterized by largeness of detail (large solid parts and large openings), the results obtained with the new formula were not as accurate as would be desired, although in general more accurate than the results obtained with the formulae and methods previously used.

2. In some cases such as ornamental grilles, heater cores and cases where deflectors or heaters are located in close proximity to the grilles (unit ventilators), there is considerable doubt as to what should be considered free area.

3. In some cases, particularly in the case of long narrow grilles, the nature of the approach results in a strip of motionless or re-entering air along one edge. Where this condition extends in for several inches from the edge, the accuracy of the results will not be seriously affected, as the anemometer will come to rest at these points and thus bring down the average reading obtained by the proper amount. In the case of a long slender grille, however, a strip of quiescent air, say ¾ in. wide, will not be sufficient to cause the instrument to read lower and yet is great enough to cause considerable error if not considered in the calculations.

5. HEATING UNITS—Radiation—Direct and Indirect.—*Technical Advisory Committee:* John Holton, *Chairman;* R. M. Conner, R. E. Daly, R. V. Frost, F. E. Giesecke, H. F. Hutzel, A. P. Kratz, J. F. McIntire, W. T. Miller, R. N. Trane.

The Society's research work on heating units is now being carried on under the direction of Professors Willard and Kratz at the University of Illinois in cooperation with the Society's Research Laboratory. Quoting from Professor Willard's report, the following is an outline of the work now under way and planned for the future:

GENERAL STATEMENT—This report presents in a very condensed and summarized form certain data pertinent to the objectives, accomplishments, conclusions, future program, and expenditures for the investigation of radiator performance from the funds provided by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the University of Illinois since April 1, 1931. It may be well to point out that the work in progress under the present agreement is, in part at least, a continuation of an investigation in this same field originally sponsored and supported by the *Institute of Boiler and Radiator Manufacturers.*

OBJECTIVES—The investigation as now administered, beginning April 1, 1931, has proceeded with two principal objectives, which are being pursued in two separate and distinct testing plants located in the Mechanical Engineering Laboratory at the University of Illinois.

(1) A continuation of the study, in our room heating testing plant, of the performance characteristics of the various types of cast iron radiators and various types of non-ferrous radiators having heating elements made of copper, brass, and aluminum assembled in enclosures acting as convectors. This plant was erected and completely equipped under the previous cooperative agreement with the *Institute of Boiler and Radiator Manufacturers.*

NOTE: These tests are all being conducted under heating service conditions in actual rooms with outside air temperatures approximately zero degrees and with inside air temperatures taken at the 30-in. level of approximately 68 F. Comparisons between the different radiators tested are then made on the basis of steam condensed per hour and the air temperature gradient between ceiling and floor levels. It should be noted especially that while the air temperature at the 5-ft or breathing line level is always recorded, it is the 30-in. level which is kept the same, at 68 F, and used as the common basis for comparison.

---

[1] Measurement of the Flow of Air Through Registers and Grilles, L. E. Davies, A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930.

[2] Measurement of the Flow of Air Through Registers and Grilles, L. E. Davies, A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931.

(2) An entirely new study of exposed and concealed ferrous and non-ferrous radiator performance in a special *warm-wall* testing booth. This study has involved the erection and operation of a test booth in accordance with the specifications contained in the "Proposed Code for Testing and Rating Concealed Gravity Type Radiation," Edition of January, 1931, which was adopted in principle at the last annual meeting of the Society.

NOTE: In this test booth the same radiators which have been tested in the room heating testing plant (see item 1) will be again tested in accordance with the "Proposed Code" for the purpose of obtaining correlation factors between the two plants with various types of radiators.

ACCOMPLISHMENTS—Published results of the entire investigation to date appear in three *Bulletins* Nos. 169, 192 and 223, and one Reprint, No. 1, of the Engineering Experiment Station of the University of Illinois, and in three professional papers before the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.[3, 4, 5]

ROOM HEATING TESTING PLANT

Altogether, about eight types of exposed and concealed cast-iron radiators, and about fifteen types of enclosures for these same radiators have been investigated and their performance characteristics determined in the room heating testing plant under actual winter service conditions. In addition to these tests, about four types of ferrous and non-ferrous concealed convector heaters or radiators have been investigated in a similar manner, and similar performance characteristics determined. These latter tests and some of the former have been made since April 1, 1931, and will be reported on at the annual meeting of the Society in January, 1932.

WARM WALL BOOTH

Two types of convector heaters with non-ferrous heating elements have been tested over a wide range of air temperatures in the *warm wall* booth. In each case, the relation between steam condensation and temperature of the air entering the heater has been established over a wide range of air temperatures. The testing work for establishing such relations cannot progress rapidly because the surrounding air temperatures are determined by conditions in the large laboratory in which the booth is located. After the validity of the results obtained in the *warm wall* booth and the necessary correction factors for deviations from standard entering air temperatures have been completely established, work can progress more rapidly since but one entering air temperature will then be required for each heater tested. Results of these tests comparing the performance of convector heaters with direct radiators in the *warm wall* booth as well as in the room heating testing plant will be reported on at the annual meeting of the Society in January, 1932.

CONCLUSIONS—From the results of tests in the room-heating testing plant several important conclusions relative to the performance of various types of radiators in heating rooms have been definitely established. (See [3, 4, 5] and A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932.)

From the results of a very limited number of tests on concealed convector heaters in the *warm wall* booth, the following observations or tentative conclusions have been drawn:

1. Based on the data so far obtained, the evidence supports the conclusion that the steam condensations obtained in the *warm wall* booth are valid, in that they are in agreement with those obtained in the room heating testing plant when run with the same temperature for the air entering the heater, or when the proper correction factor is applied to reduce the results to a common entering air temperature.
2. The results so far also indicate that over a fairly wide range of entering air temperatures the heat output of the convector heaters varies as the 1.3 power of the temperature difference between the steam and the entering air with sufficient accuracy to justify the use of this relation for the correction to standard entering air temperature if the deviations from the standard temperature are kept within reasonable limits.

FUTURE PROGRAM—It is quite apparent that one of the most important fields for future study in connection with radiator room heating has been indicated already in the Conclusions.

---

[3] Investigation of Heating Rooms with Direct Steam Radiators Equipped with Enclosures and Shields, A. C. Willard, A. P. Kratz, M. K. Fahnestock and S. Konzo, A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929.

[4] Wall Surface Temperatures, A. C. Willard and A. P. Kratz, A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930.

[5] Steam Condensation an Inverse Index of Heating Effect, A. P. Kratz and M. K. Fahnestock, A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931.

Some progress has been made in designing and assembling auxiliary apparatus for a eupatheoscope, or instrument for measuring the comfort conditions in a room as evaluated by the heat loss from a copper cylinder with the surface temperature maintained at 75 F. Some experimental work has been done to determine the most feasible arrangement of thermocouples and electrical heaters to be used in connection with this cylinder, as well as the adaptability of the recorders available. This eupatheoscope will prove a valuable supplement to the temperature gradients in the room in studying the heating effects of various types of radiators.

It is also probable that a rearrangement of the rooms in the present testing plant should receive very serious consideration. Undoubtedly, confirmatory tests using radiator and enclosure equipment already tested should be made in (1) large rooms of the same general proportions and characteristics, and (2) rooms with a single outside wall exposure instead of two outside walls. For these latter tests, the larger room is quite essential in order to secure adequate heat loss with only one wall.

In the case of the *warm wall* booth, the future testing program should include further tests on exposed and concealed cast-iron steam radiators, more tests on convector heaters with steam, and finally, a limited series of tests on all types of radiators with hot water.

6. GARAGE VENTILATION—Ventilation of Garages and Bus Terminals.—*Technical Advisory Committee:* E. K. Campbell, *Chairman;* D. S. Boyden, H. P. Gant, W. T. Jones, W. C. Randall.

The Technical Advisory Committee has given a great deal of consideration during the past year to the continuation of the study of garage ventilation. Complete plans were recently perfected for a study to be made by the Laboratory in the five-level basement garage of the 37-story Grant Building in Pittsburgh.

7. HUMAN COMFORT—Air Conditions and Their Relation to Living Comfort.—*Technical Advisory Committee:* C. P. Yaglou, *Chairman;* O. W. Armspach, W. L. Fleisher, Dr. E. V. Hill, Dr. R. R. Sayers.

Investigations under this Technical Advisory Committee have been carried on by the Society's Research Laboratory in Pittsburgh and by Professor Yaglou of the Harvard School of Public Health.

As soon as work now under way in the psychrometric chambers in the Laboratory in Pittsburgh is completed, the following two projects will be undertaken: (*a*) A study of the physiological and bacteriological effects of humidity and changes of humidity on the human organisms. (*b*) A study of physiological and bacteriological effects of cold and the effects of drafts, low temperature, etc. on the human organism.

Under the general heading of the study of vital characteristics of the atmosphere, Professor Yaglou of the Harvard School of Public Health in cooperation with the Research Laboratory has been working on the following three investigations:

a. A study of the differences in ionic content in outdoor air and expired air, and the effects of room occupancy upon ionic content.

b. De-ionizing effect of air-conditioning methods and apparatus.

c. A thorough study of diurnal and seasonal variations in atmospheric ionization, both out of doors and indoors, in Boston, and in at least three other cities having different climates. Observations to be correlated with corresponding variation in the incidence of seasonal diseases.

Items *a*, *b* and part of *c* were approved by the Research Committee early this year and are now being studied.

The Research Committee recently approved the study of subject *c* to cover at least three other localities, namely Pittsburgh, Denver, and one or two other cities on the Pacific Coast.

The Technical Advisory Committee has recommended the following five subjects for future investigation:

a. A study of the growth of organisms and insects in air-conditioned rooms and in air washers, Baudelot coolers, etc., with the object of finding a water soluble, non-toxic, element for checking or preventing growth of lower organisms.

b. The effect of space charge (ionic unbalance) on the colloidal state of certain substances (milk, yeast, glue, and other common gelatinous substances).

c. A thorough study of air requirements for ventilation both in winter and summer under customary seasonal range of room temperature and humidity. Experiments to be carried on in a conditioned chamber using an ideal laboratory system of distribution and a temperature rise of 10 to 30 F in the summer experiments.

d. A study of air distribution with the object of improving present systems, or developing new and more efficient systems.

e. Organization of a new committee on Refrigeration in Relation to Air Conditioning for the purpose of disseminating knowledge and sponsoring research.

8. OIL HEATING—Oil and Gas Burning Devices.—*Technical Advisory Committee:* L. E. Seeley, *Chairman;* R. M. Conner, P. E. Fansler, R. S. Franklin, R. V. Frost, R. C. Morgan, H. F. Tapp.

A study of methods of testing oil burner boiler combinations is being carried on under the direction of Professor Seeley of Yale University in cooperation with the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *American Oil Burner Association.* The investigation has for its purpose the development of a standard code for testing oil burners—boiler or furnace combinations. As a result of the work so far completed, there has been prepared and submitted to the Society a Code for Testing Low-Pressure Steam Heating Boilers Burning Oil Fuel.[6] Also, a complete report of a study of sixteen combinations of different boilers and burners showing (*a*) effect of various fuel burning rates upon quality of combustion (*b*) effect of draft fluctuation upon combustion (*c*) the effect of quality of combustion upon the efficiency and upon the flue gas temperature, (*d*) characteristic efficiency curves, (*e*) influence of heating surface upon efficiency, (*f*) influence of burner type on efficiency, (*g*) sound measurements, (*h*) power requirements of oil burners. (*i*) actual heat liberation rates per cubic foot of furnace volume, (*j*) characteristic efficiency curves due to intermittent burner operation, etc.

9. SOUND—Sound in Relation to Heating and Ventilation.—*Technical Advisory Committee:* F. B. Rowley, *Chairman;* Carl Ashley, Warren Ewald, R. F. Norris, G. T. Stanton.

This is a new project given consideration by the Committee on Research and the Technical Advisory Committee during the past year. No work has been undertaken in the Laboratory. The Technical Advisory Committee is studying the problem under the following three headings:

a. The selection or development of a standard method and standard equipment for measuring sound. This must be accurate, reliable and should be reasonably cheap and convenient for field operation.

b. A study of the causes of sound in ventilating equipment and a better understanding of methods of eliminating sound at its source.

c. A study of the methods of reducing the effect of unavoidable sounds and noises to unobjectionable levels.

10. AERATION—Infiltration in Buildings.—*Technical Advisory Committee:* G. L. Larson, *Chairman;* J. E. Emswiler, F. E. Giesecke, W. C. Randall, W. A. Rowe, J. G. Shodron, Ernest Szekely, M. S. Wunderlich.

With the completion of the studies of infiltration through built-up wall sections of different types of construction, the future study of infiltration has been directed towards other phases of the general question of aeration of buildings. Five projects have been approved by the Technical Advisory Committee (*a, b, c, d,* and *e*). Work on some of these projects was carried on during 1931 under the direction of Professor Larson at the University of Wisconsin in cooperation with AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Laboratory. The committee has recommended that other work proposed by it be carried on at the laboratory in Pittsburgh and elsewhere. The following are the five projects as prepared by the Technical Advisory Committee:

a. STUDY OF DOUBLE HUNG WOOD WINDOWS.

b. EFFECT OF ELIMINATING VENT OUTLETS IN SCHOOL ROOM VENTILATION, OR EXFILTRATION IN VENTILATED AREAS.

[6] See A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931.

c. AIR DRIFT AROUND BUILDINGS.

d. AIR STRATIFICATION IN CONFINED SPACES AND THE EFFECT OF CROWDS AND AIR MOVEMENT ON THE SAME.

e. INFILTRATION AND EXFILTRATION IN TALL BUILDINGS.

11. PUMPS AND TRAPS—Devices for Handling Condensation and Air.—*Technical Advisory Committee:* E. K. Lanning, *Chairman;* W. E. Barnes, C. A. Dunham, I. C. Jennings, F. J. Linsenmeyer, J. C. Matchett, A. W. Moulder, E. J. Ritchie, W. K. Simpson, H. G. Thomas, R. H. Thomas.

This is a new project considered by the Committee on Research during the past year. The technical Advisory Committee is studying the problem with a view of outlining its needs.

12. REFERENCE DIRECTORY—Correlating Thermal Research.—*Technical Advisory Committee:* R. M. Conner, *Chairman;* D. S. Boyden, P. D. Close, J. C. Fitts, W. T. Jones, H. T. Richardson, Perry West.

At the last annual meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS held in Pittsburgh, January 26 to 29, 1931, the Research Committee received a proposal from the Directors of the *Association for Correlating Thermal Research* that the card catalogs of available literature developed by the Association together with the physical property of the Association and the funds and pledges in its treasury be taken over and administered as a function of the Research Laboratory.

The Committee on Research agreed to take over the property and activities of the Association at the close of the present year, with the understanding that the Association continue to function under its Board of Directors and its own finances during the year. The Technical Advisory Committee on Correlating Thermal Research of the Committee on Research was appointed to look after the Research Laboratory's interest in the undertaking during the current year, and to receive the property and administer the work after the transfer to the Laboratory had been made. This transaction will be consummated before the annual meeting of the Society.

Financial Report of Research Laboratory

January 9, 1932.

RESEARCH LABORATORY OF THE AMERICAN SOCIETY OF
HEATING AND VENTILATING ENGINEERS,
51 MADISON AVENUE,
NEW YORK, N. Y.

Gentlemen:

As requested I audited the books of account and records of the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa., for the year ended December 31, 1931, and submit herewith my report.

CASH RECEIPTS AND DISBURSEMENTS

RESEARCH LABORATORY OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—PITTSBURGH, PA.

For the Year Ended December 31, 1931

BALANCE JANUARY 1, 1931, PER FORMER REPORT			$10,791.18
RECEIPTS			
From the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS	$22,519.85		
Other Contributions—Per Schedule	10,283.00	$32,802.85	
Interest on Bank Balances	71.21		
Interest on Securities—Research	135.00		
Interest on Securities—Research Endowment Fund	1,025.00	1,231.21	34,034.06
			44,825.24
DISBURSEMENTS			
Salaries—Per Schedule		18,517.63	
Traveling—F. C. Houghten	1,204.84		
Traveling—Staff	86.24		
Traveling—Executive Committee	397.13	1,688.21	
Laboratory Equipment and Supplies		1,131.36	
University Cooperation—Per Schedule		14,448.60	
Meetings		93.75	

OFFICE EXPENSES			
Telephone and Telegraph	68.33		
Postage	121.56		
Stationery and Printing	22.95		
Professional Services	100.00		
Expressage	9.51		
Miscellaneous	296.42		
Furniture and Fixtures	70.00	688.77	
EXPENDITURES—LABORATORY			36,568.32
OTHER EXPENSES			
Research Booklet			
Paper and Printing	391.08		
Cuts and Photos	478.86		
Preparation	200.00		
Multigraphing	16.89		
Mailing and Postage	120.51	1,207.34	37,775.66
BALANCE DECEMBER 31, 1931			$ 7,049.58

BUDGET COMPARISON

RESEARCH LABORATORY OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—PITTSBURGH, PA.

For the Year Ended December 31, 1931

	ANNUAL DISBURSEMENTS	BUDGET PROVISION	Increases *Decreases*
Salaries	$18,517.63	$19,257.00	*$ 739.37*
Office Supplies and Expense	688.77	800.00	*111.23*
Traveling	1,688.21	2,000.00	*311.79*
Laboratory Supplies and Equipment	1,131.36	2,000.00	*868.64*
Contracts with Cooperating Institutions	14,448.60	12,150.00	2,298.60
Meetings	93.75	200.00	*106.25*
Contingent		400.00	*400.00*
Promotional Booklet and Other Promotional Work	1,207.34	2,000.00	*792.66*
Stenographic Help—New York Office		800.00	*800.00*
	$37,775.66	$39,607.00	*$1,831.34*

COOPERATIVE RESEARCH PAYMENTS

RESEARCH LABORATORY OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—PITTSBURGH, PA.

For the Year Ended December 31, 1931

Armour Institute of Technology	$ 573.60
Association for Correlating Thermal Research	500.00
Harvard University	500.00
Texas A. & M. College	950.00
University of Minnesota	1,750.00
University of Minnesota (Special Heat Transmission Study)	2,500.00
University of Illinois	3,750.00
University of Wisconsin	1,375.00
Washington University	250.00
Yale University	2,300.00
	$14,448.60

Respectfully submitted,
FRANK G. TUSA,
Certified Public Accountant.

Report of Committee on Meetings Program

The activities of the Committee on Meetings Program were summarized by Prof. G. L. Larson, chairman, who suggested that the new Committee on Meetings Program endeavor to obtain papers on proportioning of heating surface in tall buildings, as there have been numerous requests for information on this subject. He also suggested papers on lag factors in heating plants showing the true relation between the boiler load and the actual load, rather than the relation between the boiler load and the connected load.

The report of the Society's representative on the Committee of Ten was submitted by J. H. Walker. A report was also submitted by T. M. Dugan relative to *American Standards Association* Committees dealing with valves, fittings and pipe threads.

Report of Guide Publication Committee

The next report was that of the Guide Publication Committee which was briefly summarized by D. S. Boyden, chairman.

STATISTICAL DATA

Number of books printed	12,000
Number of advertisers	188
New advertisers	40
Number of pages—Text Section	552
Number of pages—Advertising Section	304
Number of pages—Membership Section	64
Income from Advertising, THE GUIDE 1932	$33,260.82
Cost of Producing THE GUIDE 1932	$29,422.87

The first edition of THE GUIDE was published in 1922. Since that time many developments and changes have taken place in the field of heating, ventilation and air conditioning. The respective Guide Publication Committees have endeavored to keep pace with these developments in the successive issues.

The 10th edition of THE GUIDE has been extensively revised and amplified. The Text Section has been enlarged and the number of chapters increased from 35 to 40, more than half of the subject matter being new material.

In behalf of the members of the Society and the profession at large, the Guide Publication Committee extends its sincere appreciation for the loyal cooperation given by the following engineers whose efforts have made the Text Section of THE GUIDE 1932 the largest and most comprehensive of these annual reference volumes:

HAROLD L. ALT
O. W. ARMSPACH
J. L. BLACKSHAW
C. A. BOOTH
C. P. BRIDGES
R. M. CONNER
V. S. DAY
N. W. DOWNES
M. W. EHRLICH
J. E. EMSWILER
M. D. ENGLE
J. A. FLEMINGS
F. E. GIESECKE
F. W. HANBURGER
L. A. HARDING
ELLIOTT HARRINGTON
J. H. HOLTON
F. C. HOUGHTEN
JOHN HOWATT
A. F. KARLSON
A. P. KRATZ
G. L. LARSON
F. J. LINSENMEYER
GEORGE W. MARTIN
V. D. MILLIKEN
J. G. MINGLE
A. W. MOULDER
H. C. MURPHY
PERCY NICHOLLS
R. W. NOLAND
E. C. RACK
W. M. RICHTMANN
S. I. ROTTMAYER
C. J. SCANLAN
L. E. SEELEY
E. SZEKELY
J. H. WALKER
C. P. YAGLOU

In the Catalog Data Section will be found sizes, shapes, capacities, dimensions, space requirements and applications so important to the engineer and contractor in planning, specifying and installing materials and equipment. The two sections are so interlocked that they are practically indispensable to each other, and for the best results it is recommended that the corresponding references in each section be consulted in all cases. The Catalog Data Section is dedicated to the presentation of reliable facts and figures concerning the products shown, eliminating as far as possible unnecessary *sales talk* and comparisons.

It is the sincere hope of the Guide Publication Committee that this book will be increasingly valuable to the profession at large, and to the students in schools and colleges who have found it to be a reliable and valuable text book.

GUIDE PUBLICATION COMMITTEE

D. S. BOYDEN, *Chairman;*
W. L. FLEISHER,
H. S. HALEY,
S. R. LEWIS,
L. T. M. RALSTON,
P. D. CLOSE, *Technical Secretary.*

Report of Tellers of Election

The report of the Tellers of Election was submitted by F. A. Kitchen, chairman, as follows:

President—F. B. ROWLEY	578
First Vice-President—W. T. JONES	582
Second Vice-President—C. V. HAYNES	574
Treasurer—F. D. MENSING	582
Members of the Council—Three Year Term:	
F. E. GIESECKE	583
G. L. LARSON	583
J. F. McINTIRE	580
W. E. STARK	582
Members of the Committee on Research—Three Year Term:	
D. E. FRENCH	569
F. E. GIESECKE	568
L. A. HARDING	570
A. P. KRATZ	569
G. L. LARSON	570
Total ballots cast	644
Disqualified	59

Scattered votes were recorded for various other members.

TELLERS OF ELECTION
F. A. KITCHEN, *Chairman,*
C. E. LEWIS
R. G. OLSON

Cleveland, Ohio, January 26, 1932.

The third session was a joint meeting with the *American Society of Refrigerating Engineers,* and was held at the Little Theatre of the Cleveland Auditorium, with President Carrier in the chair.

President Carrier introduced A. H. Baer, president of the *American Society of Refrigerating Engineers,* and then announced the first paper.

The fourth session was a joint meeting with the *American Society of Refrigerating Engineers* and was sponsored by that organization, President A. H. Baer of the *A.S.R.E.,* presiding.

Mr. Baer introduced the officers of the A.S.H.V.E. who were present, namely President Carrier, Vice-President Rowley, and Secretary Hutchinson, and then announced the first paper entitled, Application of Refrigeration to Heating and Cooling of Homes, by A. R. Stevenson, Jr., F. H. Faust and E. W. Roessler, which was presented by Mr. Stevenson. In this paper the authors discussed the heat pump principle, gave an analysis of conditions in electric current costs and explained some of the difficulties involved.

The next paper on the program was entitled, Bacteria as Affected by Temperature, and was presented by the author, Dr. S. C. Prescott of the Massachusetts Institute of Technology.

The last paper was entitled, Air Conditioning as Applied to Cold Storage and a New Psychrometric Chart, by C. A. Bulkeley.

These papers were published in *Refrigerating Engineering,* the official organ of the *American Society of Refrigerating Engineers.*

President Carrier announced that the fifth session would be devoted to codes.

The first report was that of the Committee on Code for Testing and Rating Unit Ventilators and was submitted by John Howatt, chairman. This report had previously been sent to members of the Society. After a motion by G. E. Otis to the effect that the code be adopted subject to mail vote with the proviso that it be put into effect on January 1, 1934, which motion was seconded by J. J. Aeberly, the code was placed before the meeting for discussion.

Perry West offered a number of suggestions which he thought would improve the code. He suggested that the changes and corrections he had offered be included in the code before it is issued, and requested that the original motion be amended to this effect, which amendment was accepted by the mover and seconder.

J. D. Cassell inquired as to whether the amendment by Mr. West included the suggestions of Dr. Brabbée and Professor Brown. Mr. Aeberly suggested that the changes be submitted to the committee to revise the report as it deemed advisable to clarify the code.

The amendment by Mr. West was seconded and carried.

President Carrier called for the motion as it had been amended, and the motion was carried. He then instructed the committee to revise the code in accordance with the suggestions received and stated that it be submitted to the members of the Society for vote by letter ballot.

Committee on Ventilation Standards Reports

The report of the Committee on Ventilation Standards, was given by W. H. Driscoll, chairman. Before proceeding with the details of the report, Mr. Driscoll paid tribute to the personnel of the committee for their splendid, intelligent cooperation, and for the vast amount of time, effort and personal expense they had given to the matter. Mr. Driscoll also outlined the various steps leading up to the present report.

The Report of the Committee on Ventilation Standards, which had been prepared in mimeograph form and distributed to the audience, was read by Mr. Driscoll.

President Carrier stated that it was his understanding that the report would contain an appendix which would include definitions and recommendations as to the methods for bringing about the desired results, to which Mr. Driscoll replied that the only items which had been definitely decided upon for inclusion in the appendix were definitions of terms and tables of effective temperatures. He said there was also a possibility that the appendix might include a reference to natural ventilation and possibly to air volume.

In order to place the report before the meeting for discussion, J. J. Aeberly moved that the report be accepted and sent to the membership for adoption by letter ballot. This motion was seconded by Thornton Lewis.

There was an extended discussion with the following participants:—H. M. Nobis, J. J. Aeberly, M. G. Harbula, G. E. Otis, C. P. Yaglou, C. H. B. Hotchkiss, J. I. Lyle, W. L. Fleisher, C. W. Brabbée, T. M. Cunningham, F. Paul Anderson, E. Q. Cole, J. D. Cassell, Thornton Lewis and Perry West.

G. E. Otis suggested an amendment to the original motion to eliminate reference to the 10 cu ft air quantity. Mr. Driscoll said the motion was out

of order because there was a motion before the house, but President Carrier pointed out that this was an amendment to the motion. Mr. Driscoll said the motion was to be sent out for a mail ballot, and suggested that the motion as originally made should use the word *received* rather than *accepted,* that is that the *report be received and sent to the membership for vote of adoption.* He said that receiving the report at the present meeting does not accept any part of it, and that he thought it unwise to accept or amend the report in its present form until it had gone before the members of the Society with the discussions that have been submitted at the meeting, so that the members would have an opportunity to consider all angles and to vote intelligently on the report.

Mr. Otis replied that he would withdraw his motion and Mr. Aeberly accepted the corrected motion. J. D. Cassell presented a motion to amend the original motion that the report sent to the membership be accompanied by the Appendix. Mr. Driscoll said that this was intended, and Mr. Cassell therefore withdrew his amendment.

Mr. Driscoll referred to a statement to the effect that the report had not been submitted in time to give those present at the meeting an opportunity to digest it thoroughly. He said that the time required to discuss the matter of specifying 10 cu ft of air per minute per person was as much responsible for this delay as any other factor. The 10 cu ft was adopted because of the experiences of the members of the committee and such scientific data as were available, such as the figures supplied by Professor Yaglou. Mr. Driscoll emphasized the fact that the primary object of the committee was to crystallize the various ideas and viewpoints of the members and to submit them to the Society for consideration.

President Carrier called for the question. Mr. Otis withdrew the motion he had made regarding certain details of the code with the understanding that there would be an opportunity to amend the code at some future meeting. President Carrier replied that a code was always open to amendment, and Mr. Driscoll said that there was no need of rushing the matter through without giving it due consideration. He thought it was too important a matter to rush it through without giving the members of the Society an opportunity to go over the report and the discussions and to express themselves at another meeting of the Society.

Mr. Aeberly and his seconder withdrew the original motion regarding the question of the mail ballot, substituting that the report come up at the next Semi-Annual Meeting of the Society for a vote.

Dean Anderson submitted a substitute motion—that the report be received. Mr. Aeberly seconded the motion. The motion was carried.

Mr. Driscoll moved that a new committee be appointed by the incoming president of the Society to give consideration to the discussion and information presented at the present meeting, that this committee be provided with the verbatim record of the discussions, that the committee amend the report if in its judgment it is advisable to do so, and that the report as amended be printed and submitted to the membership for a discussion at the Semi-Annual Meeting 1932. The motion was seconded by Mr. Otis and carried.

A motion by L. A. Harding that the secretary be instructed to extend to Mr. Barker the appreciation of the AMERICAN SOCIETY OF HEATING AND VENTI-

LATING ENGINEERS for the presentation of his most excellent and instructive paper, was seconded and carried.

The resolutions prepared by a committee of which H. M. Hart was chairman were presented and adopted:

Resolved, that the members here assembled at the 38th Annual Meeting of the Society extend to the Cleveland Chapter and its Committee on Arrangements and its committee chairman, T. A. Weager, our congratulations on the very interesting and unique program provided for our entertainment.

Resolved, that the Ladies Committee of our 38th Annual Meeting be congratulated for the graceful and cordial manner in which they looked after the comforts of their guests.

Resolved, that the Society express its appreciation to the Chamber of Commerce of Cleveland for its generous entertainment and co-operation during this convention.

Resolved, that the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS here assembled wish to congratulate the Program Committee, the authors and the Technical Secretary of the Society, for the excellent papers prepared and presented at this, our 38th Annual Meeting.

Resolved, that it is the sense of this meeting that we express our appreciation to the City of Cleveland for the use of the Public Auditorium.

Resolved, that the Society express its appreciation to the Exposition Management for the character and magnitude of the exhibits we have been privileged to attend.

Resolved, that we express our appreciation to the press for the generous and helpful publicity given this convention.

Resolved, that the Secretary of our Society be requested to express to the management of the Hotel Statler our appreciation of the excellent accommodations provided and the service rendered during our stay in Cleveland.

Resolved, that we extend our appreciation to the several Cleveland manufacturers and the Goodyear-Zeppelin Company whose plants were thrown open for our visits of inspection.

Resolved, that the Society here represented express by a rising vote of thanks the appreciation of the membership for the fine work done for the Society by our Officers, the Secretary and his staff, Committee on Research and the Laboratory staff during the past year.

Resolutions were also offered by J. F. Hale in behalf of two recently deceased members of the Society, as follows:

Whereas, James H. Davis was a charter and life member of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, and

Whereas, Mr. Davis gave much of his time and help in the early developments of our organization, serving on committees and at one time a member of our Board of Governors, and

Whereas, he has passed on to eternal life with Infinite Mind; be it

Resolved, that we in convention assembled do bow our heads in respect for him as a man; and be it further

Resolved, that this resolution be spread upon our records and that a copy be forwarded to his widow and family, that they may know with what high regard we held him.

Whereas, J. J. Wilson was a charter and life member of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, and

Whereas, Mr. Wilson was a constant attendant at our Society and Chapter meetings since their organization, and

Whereas, this charming character has passed on to a much merited reward, be it

Resolved, that we in this convention assembled do bow our heads in respect for him who took such pleasures in our association, and be it further

Resolved, that this resolution be spread upon our records, that he may be remembered for all time as one of the faithful.

President Carrier called attention to the death of Robert Pryor, another active member of the Society, who died last June, and requested those present to bow their heads for a moment in his memory.

The final matter of business was the installation of the newly elected officers by a special committee composed of three past presidents—L. A. Harding, Buffalo; John F. Hale, Chicago; and F. Paul Anderson, Lexington, Ky. The officers elected and installed were as follows: F. B. Rowley, President; W. T. Jones, First Vice-President; C. V. Haynes, Second Vice-President; F. D. Mensing, Treasurer; and Council Members: F. E. Giesecke, G. L. Larson, J. F. McIntire, and W. E. Stark.

PROGRAM 38TH ANNUAL MEETING

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

HOTEL STATLER, CLEVELAND, OHIO

JANUARY 25-29, 1932

Monday—January 25th

8:00 A.M. Finance Committee
9:00 A.M. Committee on Ventilation Standards
10:00 A.M. Meeting of the Council (*Parlor C*)
12:00 NOON Registration (*Mezzanine*)
12:30 P.M. Luncheon for Officers and Authors (*Parlor G*)
1:30 P.M. Meeting Committee on Research (*Parlor O*)
2:00 P.M. Opening of Second Heating and Ventilating Exposition (Cleveland Auditorium)
7:00 P.M. Joint Dinner-Meeting of the Councils of the A.S.H.V.E. and A.S.R.E. and their wives (Hotel Cleveland)
8:00 P.M. Committee on Radiation (*Parlor R*)

Tuesday—January 26th

9:30 A.M. FIRST SESSION (*Ball Room,* Hotel Statler)
Address of Welcome
Response by Pres. W. H. Carrier
Reports of Officers
Report of Council
Reports of Council Committees
Membership, Roswell Farnham, *Chairman*
Publication, W. A. Rowe, *Chairman*
Report of Committee on Research, C. V. Haynes, *Chairman*
Report of Director of Research Laboratory, F. C. Houghten
Reports of Special Committees
Guide Publication Committee, D. S. Boyden, *Chairman*
Committee on Increase of Membership, C. W. Farrar, *Chairman*
Reports of Continuing and Cooperating Committees
Report of Tellers of Election, F. A. Kitchen, *Chairman*

Technical Papers:
Surface Coefficients as Affected by Direction of Wind, by F. B. Rowley and W. A. Eckley
Conductivity of Concrete, by F. C. Houghten and Carl Gutberlet
Transmission of Radiant Energy Through Glass, by R. A. Miller and L. V. Black

2:00 P.M. Second Session (*Ball Room,* Hotel Statler)
Technical Papers:
Importance of Radiation in Heat Transfer Through Air Spaces, by E. R. Queer
Heat Emission from Iron and Copper Pipe, by F. C. Houghten and Carl Gutberlet
Supplementary Friction Heads in One-Inch Cast Iron Tees, by F. E. Giesecke and W. H. Badgett
Some Fundamental Considerations of Corrosion in Steam and Condensate Lines, by R. E. Hall and A. R. Mumford

7:00 P.M. Inspection of Second Heating and Ventilating Exposition

9:00 P.M. Informal Reception including entertainment and Monte Carlo (*Ball Room,* Hotel Statler)

Wednesday—January 27th

10:00 A.M. Third Session (*Little Theatre,* Cleveland Auditorium)
Joint Meeting with *A. S. R. E.*—Program arranged by A. S. H. V. E.—Pres. W. H. Carrier presiding
Technical Papers:
Changes in Ionic Content of Air in Occupied Rooms Ventilated by Natural and by Mechanical Methods, by C. P. Yaglou, L. C. Benjamin and S. P. Choate
Acoustical Problems in Heating and Ventilating of Buildings, by V. O. Knudsen
Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw, Paul McDermott and E. M. Pugh
Field Studies of Office Building Cooling, by S. S. Sanford, E. P. Wells and J. H. Walker

1:30 P.M. Trip to Akron (Goodyear-Zeppelin Hangar) with *A. S. R. E.*
(Busses leave from Hotel Statler)

7:00 P.M. Dinner for Past-Presidents and Wives (*Tavern Room,* Hotel Statler)

8:00 P.M. Chapter Relations Committee
Ladies' Theatre Party at Keith's Palace
Inspection of Second Heating and Ventilating Exposition

Thursday—January 28th

10:00 A.M. Fourth Session (*Little Theatre,* Cleveland Auditorium)
Joint Session with *A. S. R. E.*—Program arranged by *A. S. R. E.*—Alvin H. Baer presiding
Technical Papers:
Application of Electric Refrigeration to the Heating and Cooling of Houses, by F. H. Faust, E. W. Roessler and A. R. Stevenson, Jr.
Air Conditioning Applied to Cold Storage Including a New Psychrometric Chart, by C. A. Bulkeley
Biological Aspects of Thermal Engineering, by Samuel R. Prescott

12:30 P.M. Luncheon Meeting
Nominating Committee

2:00 P.M. Fifth Session (*Ball Room,* Hotel Statler)
Report of the Committee on Code for Testing and Rating Unit Ventilators, John Howatt, *Chairman*
Report of Committee on Ventilation Standards, W. H. Driscoll, *Chairman*

2:30 P.M. Ladies' Bridge (Chamber of Commerce)

7:00 P.M. Annual Banquet and Dance (*Ball Room,* Hotel Statler)

Friday—January 29th

10:00 A.M. SIXTH SESSION (*Ball Room,* Hotel Statler)
Technical Papers:
A Study of Intermittent Operation of Oil Burners, by L. E. Seeley and J. H. Powers
Room Warming by Radiation, by A. H. Barker
Comparison of Performance of Convector Heaters, by A. P. Kratz
A Study of the Combustible Nature of Solid Fuels, by R. V. Frost
Installation of Officers
Resolutions
Adjournment

12:30 P.M. Luncheon and Meeting of Council (*Tavern Room,* Hotel Statler)

2:30 P.M. Inspection Trips to Cleveland Industrial Plants

COMMITTEE ON ARRANGEMENTS

T. A. WEAGER, *General Chairman*

Finance: Walter Klie, *Chairman;* W. E. Stark, H. M. Nobis.

Banquet: W. C. Kammerer, *Chairman;* C. W. St. Clair, Vincent Eaton.

Reception and Registration: C. F. Eveleth, *Chairman;* F. A. Kitchen, C. Gottwald.

Transportation and Inspection Trips: F. H. Morris, *Chairman;* C. J. Deex, J. E. Beyer.

Entertainment: R. G. Davis, *Chairman;* E. H. Pogalies, S. H. Givelber.

Publicity: M. F. Rather, *Chairman. Akron District*—D. E. Humphrey. *A.S.R.E.*—D. F. Keith.

Bridge Hostesses: Mrs. Walter Klie, Mrs. H. B. Matzen, Mrs. C. F. Eveleth, Mrs. T. A. Weager, Mrs. D. F. Keith, Mrs. C. W. Colby.

No. 913

A. S. H. V. E. STANDARD CODE FOR TESTING AND RATING STEAM UNIT VENTILATORS

COMMITTEE—John Howatt, *Chairman,* C. P. Bridges, S. E. Dibble, Warren Ewald, L. D. Harnett, G. E. Otis, R. C. Ott

A. OBJECT OF THE CODE

1. **The Object** of this code is to provide a method of testing and rating the heat and air output of steam unit ventilators.

B. DEFINITIONS

2. **A Steam Unit Ventilator** for the purposes of this code shall be construed to mean a combined heating and ventilating unit arranged with connections for taking air direct from outdoors and with the heating unit heated with low-pressure steam and with the fan or fans and motor as an integral part of the unit.

3. **The Entering Temperature** for the purposes of this code shall be construed to mean the average temperature in degrees Fahrenheit of the air entering the unit measured at the air inlet.

4. **The Final Temperature** for the purposes of this code shall be construed to mean the average temperature in degrees Fahrenheit of the air discharged from the unit measured at the air outlet.

5. **The Power Input** for the purposes of this code shall be construed to mean the power input to the motor at the stated fan speed and at the stated conditions of rating.

6. **The Steam Pressure** for the purposes of this code shall be construed to mean the gage pressure in pounds per square inch above the standard or normal barometric pressure of 14.7 lb per square inch (29.921 in. of mercury).

7. **Standard Air** for the purposes of this code shall be construed to mean air weighing 0.07495 lb per cubic foot [1] and having a specific heat of 0.2415.

8. **The Total Equivalent Direct Radiation (EDR)** for any condition of operation when supplied with steam at 2 lb gage pressure shall, for the

[1] This weight corresponds to dry air at 70 F or air with a relative humidity of 50 per cent at a dry-bulb temperature of 68 F, when the barometric pressure is 29.921 in. of mercury.

Report of Committee presented and adopted at the 38th Annual Meeting of the American Society of Heating and Vetilating Engineers, Cleveland, Ohio, January, 1932.

purposes of this code, be construed to mean the heat in the discharged air minus the heat in the entering air, in Btu at a specified inlet air temperature divided by 240.

9. **Surplus or Heating Equivalent Direct Radiation** for the purposes of this code shall be construed to mean difference between the total EDR at a specified inlet temperature and the EDR required to heat the air from that temperature to 70 F.

C. BASIS OF RATING

10. **Rating Factors to Be Specified.** The rating of the unit ventilator shall specify the following:

a. Final temperature at different entering air temperatures.
b. Total EDR at different entering air temperatures.
c. Air delivered by the unit in cubic feet per minute at the *standard basis of rating* with the fans operated at rated speed, with all air being blown through the heating unit and with the standard louvre and grille on the outlet.

11. **The Standard Basis of Rating** shall be as follows:

a. Dry saturated steam at a temperature at the unit corresponding to an absolute pressure of 16.7 lb per square inch (218.5 F).
b. Entering air temperature of zero degrees Fahrenheit.
c. Volume delivered in cubic feet per minute converted to standard air at 70 F.

12. **Rating Tables** for unit ventilators shall contain the following data in addition to the standard rating, for entering air temperatures from —30 F to +60 F:

a. Inlet temperature, degrees Fahrenheit.
b. Final temperature, degrees Fahrenheit.
c. Total EDR at the specified entering temperature.
d. Surplus or heating EDR at the specified entering temperature.

13. **Tolerance.** As there are errors of measurement and inequalities of manufacture, a variation of 2½ per cent in test results shall not be considered excessive.

D. OUTLINE OF TESTS

14. **Heat Output—Air Volume Tests.** This code prescribes tests to determine the heat output in Btu from the weight of condensation, and the volume of air in cubic feet per minute from condensation and the temperature rise of the air passing through the unit, and further prescribes a means for correcting the heat output and temperature rise as obtained under test conditions to any other condition of entering air temperature and steam pressure.

15. **Air Quantity Check Test.** A check of the air quantity determined by means of the heat content of the condensation and the temperature rise of the air passing through the unit shall be made by direct measurement with a calibrated nozzle. The results of the two methods shall agree within 5 per cent before the tests shall be considered correct. However, the results obtained by means of the condensation-temperature rise method shall govern for the purpose of rating under this code.

16. Receiving Chamber. The unit being tested shall discharge into a receiving chamber (Fig. 1) in which the air shall be thoroughly mixed in order that the true average temperature of the air may be determined. The conditions within this chamber shall be controlled in such a way that the unit will deliver the same capacity as when under normal conditions of free delivery.

17. Free Delivery Conditions. In order to make certain that the tests on

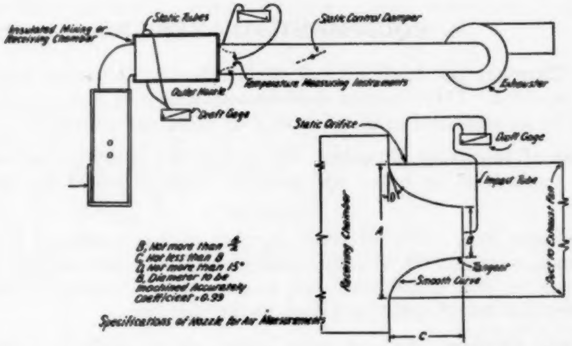


FIG. 1. RECEIVING CHAMBERS FOR TYPICAL UNIT VENTILATORS

the receiving chamber are a faithful reproduction of free delivery conditions the following tests shall be conducted:

- *a.* The condensation of the unit shall be measured during a test under actual free delivery conditions before the unit is connected with the receiving chamber.
- *b.* Three tests shall be made with the unit connected to the receiving chamber, one at a constant pressure of −0.05 in., one at a constant pressure of zero and one at a constant pressure of +0.05 in. water column on the chamber pressure draft gage.

During these tests it shall be necessary to measure only the condensation, entering air temperature and chamber pressure. These shall each be of but ½ hour duration. These four tests shall be made at the same fan speed and at entering air temperatures within 4 deg of each other.

18. Correction for Entering Temperature. The condensation measured on these three chamber tests shall be corrected to the entering temperature of the free delivery test in the proportion of the temperature difference between the steam and entering air.

19. Condensation—Chamber Pressure Curve. A curve shall then be plotted from the three tests on the chamber showing condensation against chamber pressure. This curve shall be assumed to be a straight line within this narrow range of chamber pressures. From this curve shall be read the chamber pressure corresponding to the condensation measured on the free delivery test. The draft gage, which records the receiving chamber pressure, shall be held constant at the reading thus determined, during the tests

necessary for rating data as described hereinafter. This series of tests for the calibration of the receiving chamber to produce free delivery conditions shall be made for each unit at rated speed. Should the free delivery condensation in any series of tests meet the chamber pressure curve for that series at a point above +0.05 or below —0.05 it is an indication that too great an error has been made either in the free delivery test or the check tests on the chamber, which shall be corrected or the test repeated.

E. EQUIPMENT FOR TESTING

20. **A Chamber** for receiving and mixing the air discharged from the unit shall be provided. This chamber shall be constructed of any suitable material and shall be air-tight and insulated with 2 in. cork, or equivalent.

21. **Size of Receiving Chamber.** This receiving chamber shall be of such size that the unit to be tested will produce from 20 to 90 air changes per minute.

22. **Exhaust Fan.** This receiving chamber shall be connected by a duct to an independent exhaust fan of such capacity that it will overcome the resistance of the chamber and connections and produce a zero static pressure at the point where the heater outlet is joined to the chamber.

23. **Static Orifices.** Two or more static orifices shall be located in the receiving chamber and not in the direct air blast from the unit. These openings shall be connected by air tight tubes to a common draft gage which can be read to 0.005 in.

24. **Control of Exhaust.** Means shall be provided with which to vary the capacity of the exhaust fan so as to maintain a zero static constant on the draft gage.

25. **A Calibrated Nozzle** shall be fitted into one wall of the chamber, discharging into the duct leading to the exhaust fan. The outlet opening of this nozzle shall be of such area that the air velocity is not less than 3000 fpm.

26. **Instruments** shall be located at the point of 3000 fpm minimum velocity for measuring the final temperature, which shall be the average of temperatures taken simultaneously at at least two points in the plane of the nozzle outlet for each square foot of outlet area, but in no case less than four points.

27. **A Draft Gage** shall be provided for measuring pressures at the nozzle. One side of the draft gage shall be connected to a static orifice located flush with the inner wall of the exhaust duct near the chamber wall. The other side shall be connected to an impact tube arranged to measure the velocity pressure of the air in the nozzle outlet.

28. **The Air Handled** by the unit shall be disposed of in such a way as to prevent fluctuation in the temperature of the air entering the unit.

29. **Steam Shall Be Supplied** from a source of sufficient capacity to prevent sudden changes in pressure. The pressure in the supply line up to the throttling valve shall be maintained at approximately 10 lb.

30. The Piping Diagram (Fig. 2) shows the connections and equipment prescribed for supplying steam to the coil and measuring the condensate. All of the fittings and instruments shown shall be used and shall be installed in the relative positions indicated.

31. Separators, Steam Throttle Valves, Manometer. Separators shall be of liberal capacity. Steam throttle valves shall be of a type suitable for close control. In selecting the type of manometer consideration shall be given to the fact that condensation will collect above the mercury on the steam pressure side and lend to the error unless compensated for.

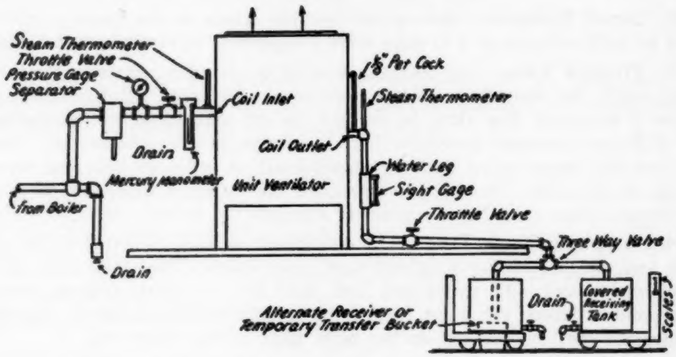


Fig. 2. Piping Diagram for Unit Ventilator Test Equipment

32. The Petcock used for air relief shall be not more than ⅛ in. in size.

33. The Water Leg shall have a sight gage so that the water level can be brought to the same point at the time of each reading of the condensate.

34. The Fittings and Piping from the radiator outlet to the thermometer shall be insulated.

35. The Scale for weighing condensate shall be of the beam type capable of being read to 0.25 lb.

36. The Tanks receiving the condensate shall be covered to reduce the loss by evaporation.

37. Temperature Measuring Instruments shall be placed around the intake of the unit in such locations and in such numbers as will reflect a true average of the entering air temperature. All temperature measuring instruments shall be capable of being read to 0.5 deg F or closer and shall be calibrated. When exposed to radiant heat they shall be shielded therefrom.

38. The Heater Casing need not be insulated for the purpose of this code.

39. A Stop Watch shall be used for the accurate timing of readings.

40. A Barometer shall be provided to determine the atmospheric pressure during test.

F. TEST PROCEDURE

41. Preliminary. The steam shall be turned on; all valves and air vents shall be opened wide for a sufficient length of time to blow out all water and air; the fans shall be started; the control shall be set to give the predetermined static pressure in the receiving chamber; the air relief cock shall be set only so that a thread of steam escapes continuously; the draft gage zero shall be checked by disconnecting the static tube; the exhaust damper shall be reset if necessary; the unit shall be warmed until conditions have become stabilized before starting test.

42. Steam Pressure. During the test the steam at the heating unit inlet shall be held constant at 2 lb gage with a superheat of not less than 2 deg.

43. Throttle Valve. An operator shall be in constant control of the throttle valve using the manometer as the reference instrument. Before the test is begun a reference line shall be marked on the manometer corresponding to the 2 lb test pressure corrected for any water in the manometer. During the test the steam valve shall be manipulated in order to hold the mercury steady at this line. In order to introduce the required superheat there shall not be less than 5 lb drop in pressure through the throttle valve.

44. Weighing Tank Tare. The condensate shall be diverted to the waste tank and the tare on the weighing tank determined. The scales shall be free in operation and both scales and tank shall be free from external contacts. The point at which the water level is to be held at each point of reading the condensate shall be marked on the sight gage of the water leg.

45. Starting Test. The steam pressure and temperature shall be checked; the time shall be noted to the second, and the water diverted to the weighing tank—this begins the test.

46. Duration of Test. The test should be continued for one hour during which time all conditions shall be held as nearly constant as possible.

47. The Following Readings shall be taken and recorded at intervals of 10 minutes or less:

a. Steam pressure
b. Steam temperature (at coil inlet and outlet)
c. Air inlet temperatures
d. Air outlet temperatures
e. Weight of condensate for the period
f. Static pressure at unit discharge
g. Watts input to motors
h. Differential pressure at the nozzle

48. Weighing of Condensate. Exactly at the end of each period the condensate shall be diverted to an alternate weighing tank or change bucket. Accumulation of condensate for the period shall be weighed and recorded.

49. Consistent Data. If the data recorded for successive periods are inconsistent or vary beyond a reasonable margin, the test shall be continued until one hour of consistent data are recorded. Care shall be exercised to bring the water level in the sight gage to the original position at each reading and to divert the condensate at the end of each reading period to the next period.

G. COMPUTATION OF RESULTS

50. Chamber Calibration Tests. The condensation obtained on each of the receiving chamber check tests (C) shall be converted by the following formula to its approximate equivalent (C') at the entering air temperature (t'_e) of the test made at the same fan speed under free delivery conditions:

$$C' = C \frac{t'_s - t'_e}{t_s - t_e} \tag{1}$$

where

t_s = the steam temperature of the receiving chamber test
t_e = the entering air temperature of the receiving chamber test
t'_s = the steam temperature of the free delivery test

51. Check Test of Air Volume. The volume of air, in cubic feet per minute, passing through the nozzle at the final temperature as of test conditions, shall be calculated by the following formula:

$$\text{Volume} = 1096\, AK \sqrt{\frac{VP}{W}} \tag{2}$$

where

A = the area of the nozzle in square feet
VP = the differential pressure across the nozzle
W = the weight of air at the temperature in the nozzle and the barometric pressure of the test
K = the coefficient of the nozzle

52. Rating Tests. The following result directly from the test data:

t_e = average temperature of entering air, degrees Fahrenheit
t_l = average temperature of leaving air, degrees Fahrenheit
t_r = temperature rise of air, degrees Fahrenheit
t_s = saturated temperature of steam, degrees Fahrenheit
p = pressure of steam, pounds per square inch, gage
b = barometric pressure, pounds per square inch
P = total or absolute pressure of steam = $p + b$
C = pounds of condensation per hour
h_{fg} = latent heat of steam

53. Rating Under Stated Conditions. It is generally impracticable to test units under exact predetermined entering air temperatures and steam pressures. Since, however, it is necessary to rate them under stated conditions, the data obtained from the test may be used for the determination of performance under such desired conditions of rating. For this purpose, the assumptions in Paragraphs 54 and 55 shall be made.

54. Constant Air Volume Assumption. It shall be assumed that the volume of air handled by the fan or fans at the temperature in the fans is constant for a given unit and fan speed regardless of temperature and barometric pressure changes.

55. Temperature Rise Assumption. It shall be assumed that the formula for determining the average temperature rise of the air under free delivery conditions, namely:

$$t'_r = \frac{t_r\,(t'_s - t'_e)}{(t_s - t_e)} \tag{3}$$

is true for a constant volume of air as well as for constant weight within the range and limits of this code.

56. Formulae for Computing Results. The following formulae may be used to compute results:

$$Q = \frac{h_{fg} \times C}{14.5 \times t_r \times W_e} \tag{4}$$

$$Q' = Q\,\frac{(460 + 70)}{(460 + t'_e)} \tag{5}$$

$$t'_r = t_r\,\frac{(t'_s - t'_e)}{(t_s - t_e)} \tag{6}$$

$$H = 14.5\,Q \times t'_r \times W'_e \tag{7}$$

where

Q = volume of air delivered, cubic feet per minute measured at inlet temperature and barometric pressure of the test
Q' = rated volume of standard air delivered, cubic feet per minute.
C = pounds of condensation per hour
h_{fg} = latent heat of steam
W_e = weight of air per cubic foot at t_e and at the barometric pressure of the test
W'_e = weight of air per cubic foot at t'_e and at standard barometric pressure (29.921 in. of mercury)
14.5 = 60 x 0.2415 (specific heat)
H = heat output, Btu per hour

57. Example. Assume the following data as having been obtained from a test: $C = 90$ lb; $t_e = 30$ F; $t_f = 111$ F; $t_r = t_f - t_e = 81$ F; barometric pressure of the test = 29.00 in. of mercury; $t_s = 219$ F; $h_{fg} = 966$ Btu per pound; rpm = 490. Substituting the proper values in Formulae 4, 5, 6 and 7:

$$Q = \frac{966 \times 90}{14.5 \times 81 \times 0.07858} = 942 \text{ cfm}$$

$$Q' = 942\,\frac{460 + 70}{460 + 0} = 1085 \text{ cfm (for } t'_e = 0)$$

$$t'_r = \frac{81\,(219 - 0)}{219 - 30} = 94 \text{ F (for } t'_e = 0)$$

$$H = 14.5 \times 942 \times 94 \times 0.08636 = 110{,}900 \text{ Btu per hour.}$$

58. Sample Table. A complete performance table may be calculated for different entering temperatures in accordance with Paragraph 57, as follows:

t_e	t_r	Btu	Total EDR	Surplus EDR
−10	98	118,300	493	91
0	94	110,900	462	118
10	90	103,900	433	114
20	85	96,200	401	165
30	81	89,700	374	190
40	77	83,600	348	213
50	72	76,600	319	230
60	68	71,000	296	253

No. 914

SURFACE COEFFICIENTS AS AFFECTED BY DIRECTION OF WIND

By F. B. Rowley [1] *(MEMBER)*, and W. A. Eckley [2] *(NON-MEMBER)*, Minneapolis, Minn.

This paper is the result of research conducted at the University of Minnesota in cooperation with the A. S. H. V. E. Research Laboratory

IN THE previous work at this laboratory on surface coefficients, the apparatus was arranged so that the direction of the wind flow was parallel to the surface of the material under test. In practice, the wind may blow at any angle to the exposed surface, and the question arises as to the relation of the surface coefficients for different angles of incidence between the wind and the surface.

Apparatus and Test Procedure

In order to determine this relation, test apparatus as shown in Figs. 1, 2 and 3, was set up and a series of tests made with wind velocities varying from 0 to 30 mph and at angles to the test surface varying from 0 deg to 90 deg. The apparatus consisted essentially of a 30-in. air duct 25 ft long, supplied with air from a variable speed fan at velocities varying from 0 to 30 mph. The air duct was provided with a Pitot tube and draft gage at a point 75 in. from the outlet end for measuring the air velocities. The test surfaces were 15 in. square and were placed with the center line 12 in. in front of the outlet end of the duct. Fig. 1 shows the fan, the 30-in. duct, and the Pitot tube arrangement with gages. Fig. 2 shows the open end of the 30-in. duct with the test specimen in place, together with the rheostat for controlling the temperatures of the test surface.

Fig. 3 shows a plan view of the outlet end of the air duct, together with a partial sectional view of the test surface and heat meter. The test surface proper was 15 in. square, with a 12-in. wing or extension wall on the leading

[1] Professor of Mechanical Engineering and Director of Experimental Engineering Laboratory, University of Minnesota.
[2] Research Assistant, University of Minnesota.

Presented at the 38th Annual Meeting of the American Society of Heating and Ventilating Engineers, Cleveland, Ohio, January, 1932, by F. B. Rowley.

side in order to direct the wind over the surface and to prevent the disturbing eddy currents from the leading edge of the test plate.

The test surface was placed against a meter which was 12 in. square and which was supplied by heat with an electrically heated plate and rheostat. The meter was similar to the Nicholl's heat meter and was the same one used in previous tests reported in the A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 429. As shown in Figs. 2 and 3, the meter was placed on a pedestal and so arranged

FIG. 1. VIEW OF FAN, 30-IN. DUCT, AND PITOT TUBE STATION

that it could be rotated on its vertical axis at any angle to the direction of the wind. The wind velocity from the duct could be varied from 0 to 30 mph; therefore, a wide range of test conditions was possible.

The differential temperatures of the heat meter, the surface temperatures of the test specimen, and the air temperatures were taken with copper constantan thermocouples and a potentiometer. The surface temperatures were taken by 28-gage copper constantan thermocouples flattened out and cemented to the surface of the test specimen with thin vellum paper. The air temperatures were taken by a thermocouple placed 1½ in. in front of the test surface. The air velocities in the duct were taken with a Pitot tube 75 in. from the outlet end, and again, by the Pitot tube placed close to the test surface to determine

the velocities parallel to the test surface, and also, the static pressures of the air at the surface. In making the tests, plate glass and smooth pine surfaces were used, and tests were made at angles varying from 0 to 90 deg through 15 deg increments. For each angle, the velocity in the air duct was varied from 0 to nearly 30 mph.

In the set-up as made, it was impossible to vary the air temperature through any wide range. Since the object of the test was to find the relation between

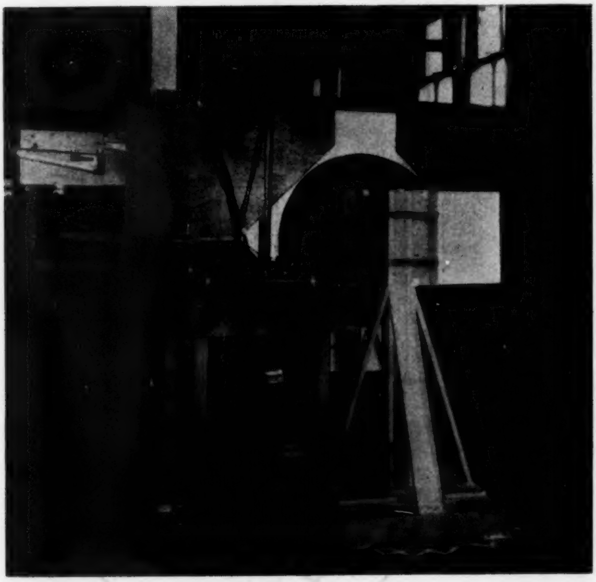


FIG. 2. VIEW OF OUTLET OF 30-IN. DUCT SHOWING TEST SURFACE AND APPARATUS

the coefficients at different wind velocities, a mean temperature was selected which was within the range of the apparatus, and this was approximated throughout all the tests. The temperature of 83 F was maintained throughout most of the tests, although in some cases there was a variation in either direction as high as 5 deg in the mean temperature. This variation was not sufficient to make any practical difference in the results. The mean temperature was taken as the average between the test surface temperature and the surrounding air temperature, it being assumed that the surrounding objects were of the same temperature as the air.

RESULTS OF TESTS

The results of the tests are shown graphically in the curves of Figs. 4 and 5. By comparing these curves with those obtained in previous tests for parallel

flow, it was found that the coefficients at zero velocity are substantially the same. At 15 mph, the coefficients for both pine and glass are slightly greater, and at 30 mph the coefficient is approximately 10 per cent greater for glass and 17 per cent greater for pine, the essential difference in the two sets of

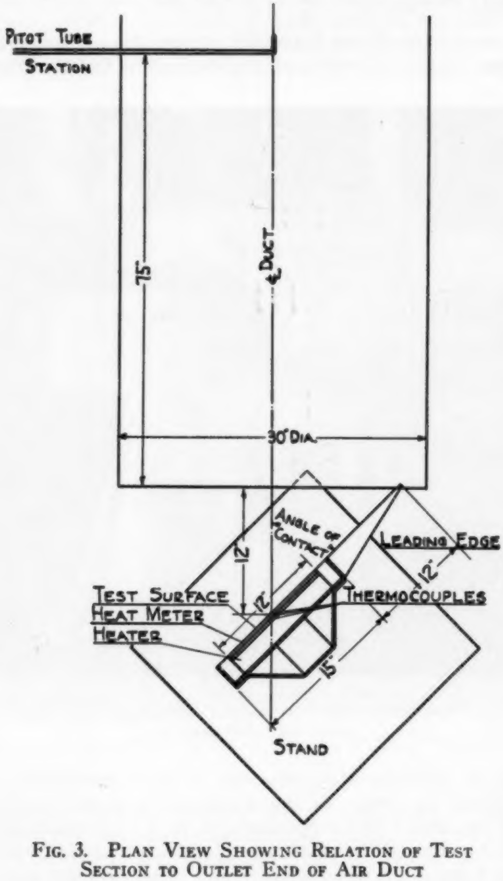


FIG. 3. PLAN VIEW SHOWING RELATION OF TEST SECTION TO OUTLET END OF AIR DUCT

curves being that, in the previous set of tests, the coefficients do not follow a straight line but as the wind velocity increases, the rate of increase for the coefficients is diminished, whereas in the present tests, the curves are substantially straight lines.

As the angle at which the wind struck the test surface was changed from zero, the coefficient was slightly reduced. For wind velocities up to 15 mph the coefficients were substantially the same for angles of 15 deg to 90 deg, all

TABLE 1. AIR VELOCITY AND STATIC PRESSURE AT SURFACE FOR VARIOUS ANGLES OF INCIDENCE OF WIND TO SURFACE

Angle of Incidence (Degrees)	Wind Velocity (Miles per Hour)	Velocity Pressure in Duct (In. of Water)	Velocity Pressure Parallel to Surface	Wind Velocity Parallel to Surface	Static Pressure at Surface (In. of Water)
	10	0.0465	0.0420	9.50	0.000
0	15	0.1045	0.072	12.46	0.004
	20	0.185	0.170	19.12	0.010
	25	0.290	0.310	25.85	0.019
	10	0.0465	0.052	10.59	0.004
15	15	0.1045	0.089	13.83	0.007
	20	0.185	0.192	20.35	0.011
	25	0.290	0.320	26.25	0.016
	10	0.0465	0.055	10.85	0.016
30	15	0.1045	0.1045	15	0.032
	20	0.185	0.190	20.22	0.054
	25	0.290	0.308	25.75	0.085
	10	0.0465	0.0465	10	0.030
45	15	0.1045	0.1045	15	0.068
	20	0.185	0.185	20	0.120
	25	0.290	0.306	25.67	0.179
	10	0.0465	0.032	8.3	0.042
60	15	0.1045	0.064	11.74	0.096
	20	0.185	0.112	15.5	0.182
	25	0.290	0.192	20.3	0.267
	10	0.0465	0.013	5.29	0.050
75	15	0.1045	0.024	7.19	0.115
	20	0.185	0.042	9.51	0.195
	25	0.290	0.076	12.8	0.309
	10	0.0465	0.002	2.07	0.053
90	15	0.1045	0.004	2.94	0.125
	20	0.185	0.009	4.4	0.230
	25	0.290	0.020	6.57	0.360

being less than for parallel flow. Above a 15-mph velocity, the coefficients were reduced as the angle was increased. On the whole, the reduction in the numerical value of the coefficient was not as much as was anticipated, and for practical purposes, the coefficients as obtained for parallel flow would be satisfactory.

In order to show the action of the air on the surface for the different angles of wind to surface, readings were taken to determine the velocity and static

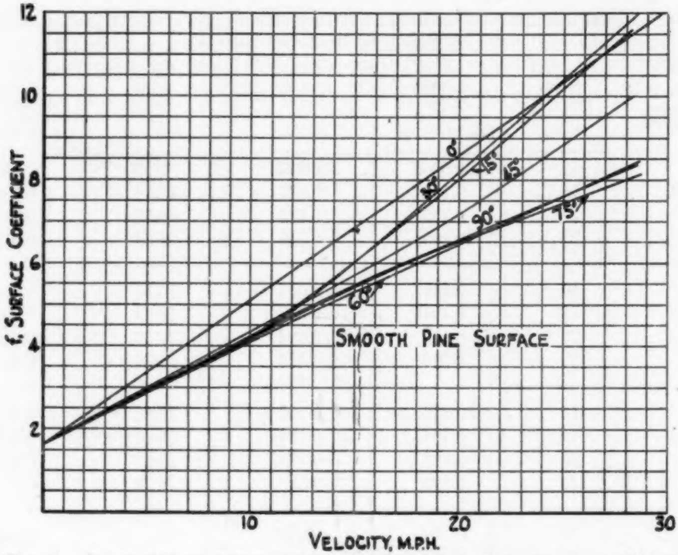


FIG. 4. SURFACE COEFFICIENTS FOR A SMOOTH PINE SURFACE FOR A WIND VELOCITY VARYING FROM 0 TO 30 MPH AND A WIND DIRECTION VARYING FROM 0 DEG TO 90 DEG TO TEST SURFACE

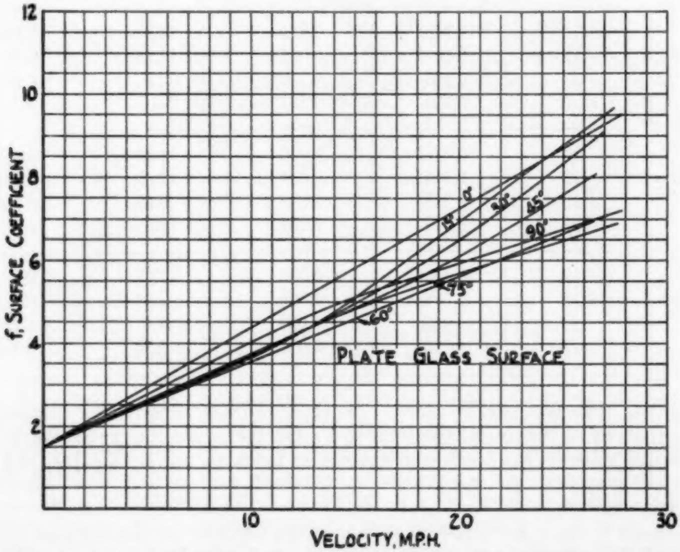


FIG. 5. SURFACE COEFFICIENTS FOR A PLATE GLASS SURFACE FOR WIND VELOCITIES VARYING FROM 0 TO 30 MPH AND WIND DIRECTION VARYING FROM 0 DEG TO 90 DEG TO TEST SURFACE

pressure of the air near the test surface for different angles and for different velocities in the air duct. The results of these readings are shown in Table 1. From these data, it will be observed that, as the angle of incidence is increased, the static pressure is gradually increased at the surface until at 60 deg the

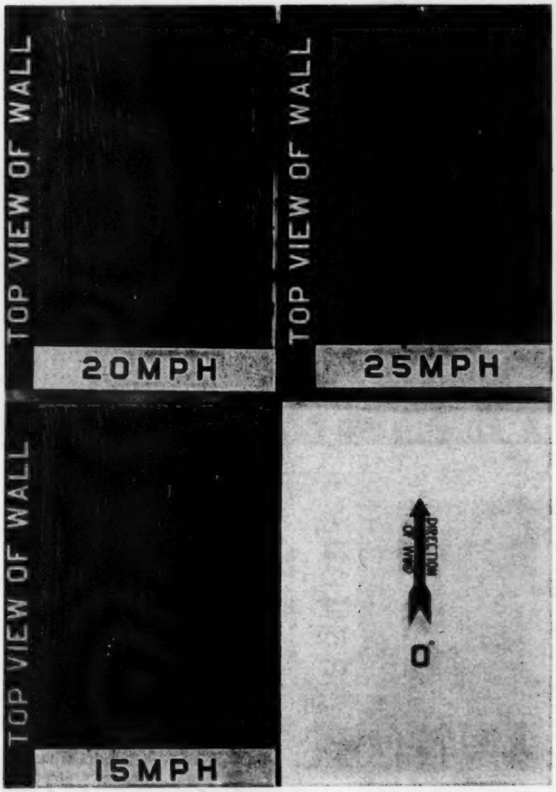


FIG. 6. LINES SHOWING DIRECTION OF AIR CURRENTS FOR A DISTANCE OF 12 IN. FROM TEST SURFACE IN A PLANE PARALLEL TO WIND DIRECTION. ANGLE OF WIND TO TEST SURFACE 0 DEG

static pressure practically equals the velocity pressure in the main duct. At 75 deg and 90 deg, it slightly passes the velocity pressure. Further, as the angle of incidence is increased, the velocity pressure on the surface, and therefore the surface velocity, are substantially the same as the velocity of the air in the duct until an angle of 45 deg is reached, after which the surface velocity is gradually reduced until it reaches a minimum at 90 deg.

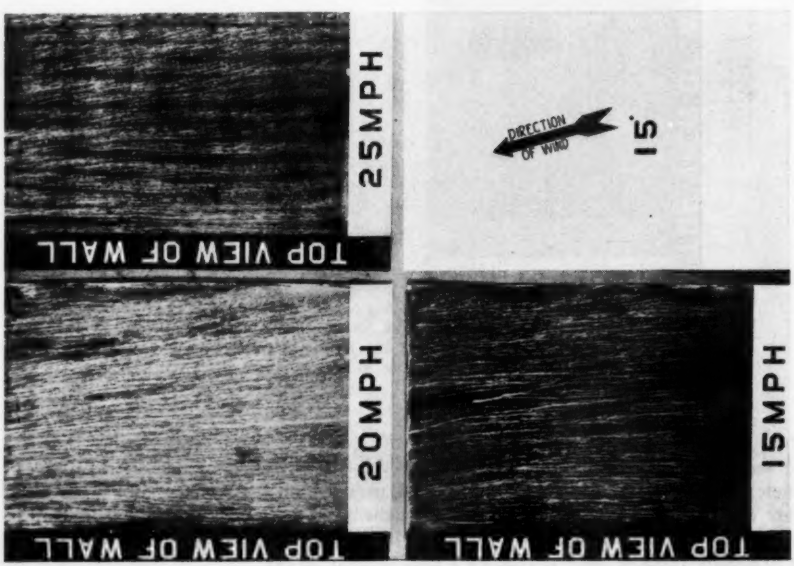


FIG. 7. LINES SHOWING DIRECTION OF AIR CURRENTS FOR A DISTANCE OF 12 IN. FROM SURFACE IN A PLANE PARALLEL TO WIND DIRECTION. ANGLE OF WIND TO TEST SURFACE 15 DEG

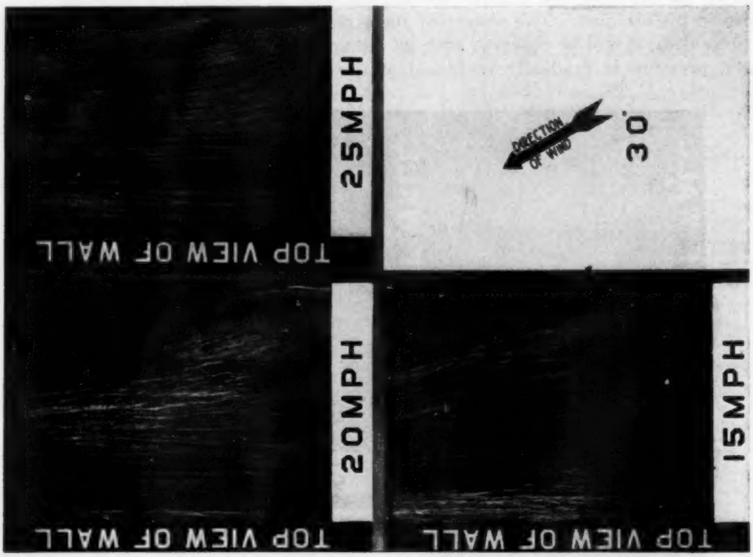


FIG. 8. LINES SHOWING DIRECTION OF AIR CURRENTS FOR A DISTANCE OF 12 IN. FROM TEST SURFACE IN A PLANE PARALLEL TO WIND DIRECTION. ANGLE OF WIND TO TEST SURFACE 30 DEG

Since the surface velocity is very greatly reduced for the high angles of incidence, it might be assumed that the surface coefficients for these conditions should also be greatly reduced. The fact that they are not is probably due to

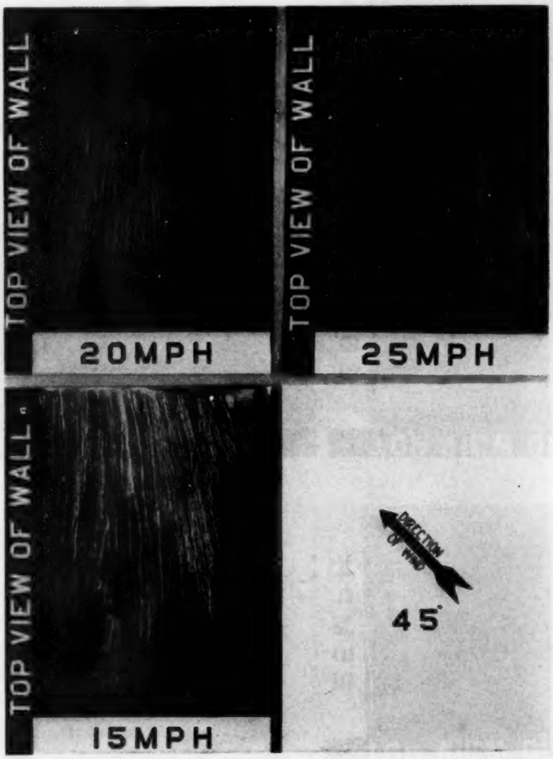


Fig. 9. Lines Showing Direction of Air Currents for a Distance of 12 In. from Test Surface in a Plane Parallel to Wind Direction. Angle of Wind to Test Surface 45 Deg

the corresponding increase in air pressure at the surface, which makes the contact between the air and surface more effective in removing the surface heat.

As a further study to determine the action of the air on the test surface for the various angles of incidence between the air and the surface, sheet metal plates, 12 in. x 18 in. in area, were placed perpendicular to the test wall and in the plane of the air flow. In this position, the plates did not disturb the air

FIG. 10. LINES SHOWING DIRECTION OF AIR CURRENTS FOR A DISTANCE OF 12 IN. FROM TEST SURFACE IN A PLANE PARALLEL TO WIND DIRECTION. ANGLE OF WIND TO TEST SURFACE 60 DEG

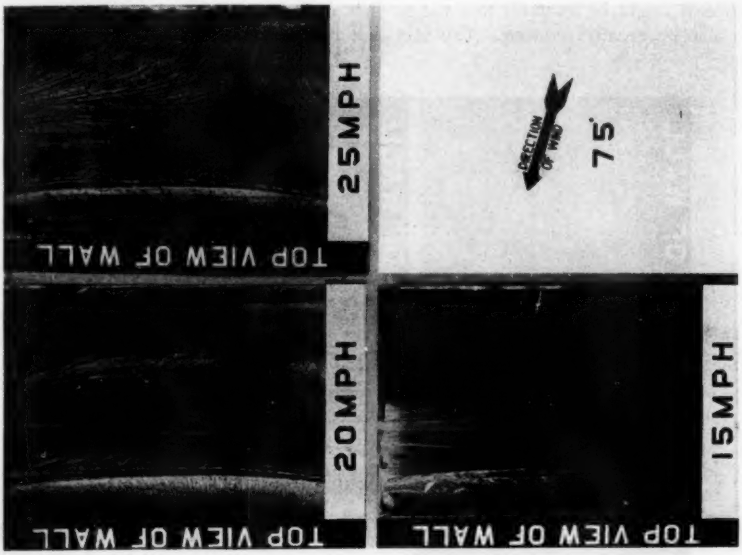


FIG. 11. LINES SHOWING DIRECTION OF AIR CURRENTS FOR A DISTANCE OF 12 IN. FROM SURFACE IN A PLANE PARALLEL TO WIND DIRECTION. ANGLE OF WIND TO TEST SURFACE 75 DEG

flow but merely separated it as it approached the wall. A light coating of lamp black and kerosene was placed on the surface of the metal sheets and the air was blown on to the test surface for a sufficient length of time to impress or mark the direction lines of the air on the surface of the plates. The plates as

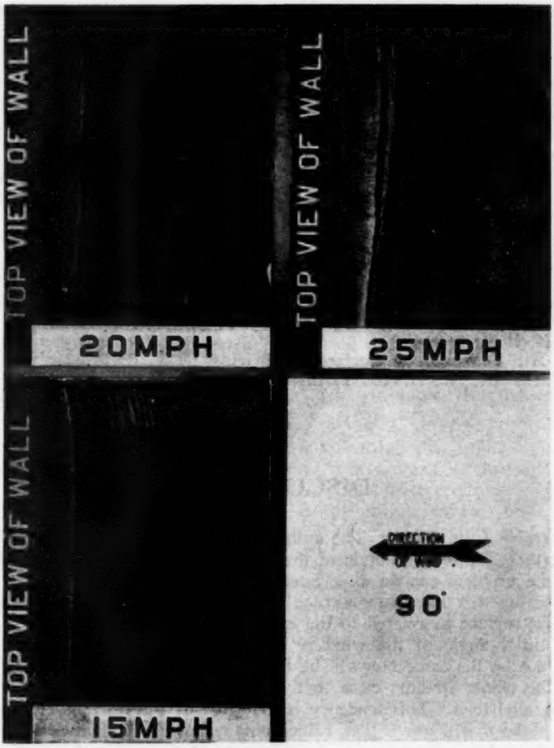


FIG. 12. LINES SHOWING DIRECTION OF AIR CURRENTS FOR A DISTANCE OF 12 IN. FROM TEST SURFACE IN A PLANE PARALLEL TO WIND DIRECTION. ANGLE OF WIND TO TEST SURFACE 90 DEG

obtained in these tests are shown in Figs. 6 to 12, inclusive. These results show very clearly the change in surface velocity conditions which take place after passing the 45 deg angle. At angles of 60 deg, 75 deg and 90 deg, it is very evident that the surface velocity is retarded but that the surface pressure is building up as was indicated by the pressure gage.

SUMMARY AND CONCLUSIONS

The results of these tests are significant, *first,* as an indication of what takes place near the surface of a wall as the angle of incidence between the surface and the wind is changed, and *second,* in that, even though there is a changed condition at the surface, the combined effect of this changed condition is such as to make the surface coefficient substantially the same as that obtained for parallel air flow. For all practical purposes, it would appear that the surface coefficients as obtained for air flow parallel to the surface might well be used without any correction for the angle of the wind. Such a correction would obviously be difficult to make, and due to the many other uncertainties surrounding this part of the problem, it would not seem justifiable to go to any such refinement. Tests were made for only one mean temperature, but there is no reason to believe that similar relations would not hold for other mean temperatures.

REFERENCES

Wind Velocity Gradients Near a Surface and Their Effect on Film Conductance, by F. C. Houghten and Paul McDermott, A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931.

Surface Conductances as Affected by Air Velocity, Temperature and Character of Surface, by F. B. Rowley, A. B. Algren and J. L. Blackshaw, A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930.

Effects of Air Velocities on Surface Coefficients, by F. B. Rowley, A. B. Algren, and J. L. Blackshaw, A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930.

DISCUSSION

P. NICHOLLS[3, 4] (WRITTEN): The authors undertook a difficult task in attempting to determine the effect of the angle of the wind on heat transfer. Personally, I fail to see how the problem can be dissociated from the area of the surface. The data obtained by using a 1 ft square surface must be limited to that surface, or more strictly to that surface as related to the extra blank foot square attached to it. Although the initial angle of the wind is known, yet its angle where it strikes the surface will not be the same but will be fixed by the resulting motions of particular portions of the whole stream; close to the surface the stream flow will, in general, tend to be parallel to it. This tendency is indicated by the tracings on Figs. 8 to 12. I think the authors will agree that values obtained could not be applied to a large area, although probably there would be the same tendency for the coefficient to decrease with increase in angle, but the effect would vary over the area.

The results show that the surface coefficient tends to be somewhat smaller with increase in angle; this conclusion agrees with natural logic for the surfaces used—wood and glass. In the tests previously reported on flow parallel to the surface, brick and stucco surfaces were included. It is quite possible that if these materials had been used with the perpendicular flow, then the surface coefficient would have shown an increase because the wind would have penetrated further into the surface and would have increased its apparent coefficient. This effect was strongly marked with pipe insulations. In service there would also be the possibility of increased infiltration through the whole wall and it would be difficult, if not impossible, to sepa-

[3] Published by permission of the Director, U. S. Bureau of Mines. (Not subject to copyright.)
[4] Supervising Fuel Engineer, U. S. Bureau of Mines, Pittsburgh, Pa.

rate the 3 components of true surface coefficient, partial penetration of air, and total penetration through the wall.

Although the authors have reasoned that the true surface coefficient is not much affected by the general direction of the wind striking it, yet I believe it would be incorrect to conclude that the total heat loss by that wall would not usually be increased by direct impingement.

Perry West: Professor Rowley clears up one point of possible controversy about transmission factors having been derived with parallel air flows and as to whether they would be the same at various air flows and at various angles.

L. A. Harding: A point that may have caused confusion to many engineers is that the impingement of air at an angle on a surface might produce such a turbulence if the flow were not parallel.

Perhaps it will be noted that the coefficients as given in The Guide for outside conditions compared with still air conditions remain as printed, that is, a ratio of 3 to 3½. It is gratifying to me to learn that our work at the University of Illinois 16 years ago produced identical results with parallel flow.

J. G. Shodron: Professor Rowley has again enriched our knowledge of certain phenomena by giving the Society more exact coefficients for surface conductance as affected by wind. Researches and reports like this one open up new vistas.

I would like to suggest further research in the field of heat emission by combining transmission losses with infiltration losses.

Yesterday I hurriedly prepared sketches in order to more clearly present two questions.

One figure shows the temperature gradient for a homogeneous wall, no air inleakage. First, what is the effect of dry air inleakage upon the outside surface coefficient and then what would the effect be on the internal conductance coefficient? When the conditions are as indicated what is the effect on the inside surface coefficient, also, on the overall transmittance coefficient?

Then considering the subject from the inside, what is the effect of humid air outleakage, is it the same as before, on the outside surface coefficient, and on the internal conductance; and on the inside surface coefficient and then on the overall transmittance?

I have noticed that the data worked up in the Laboratory does not always hold true out in the field, especially for wet walls of a porous nature, when the wind is high and cold.

Professor Rowley: The last question requires some thought. If the infiltration didn't reduce the temperature drop between the two walls, I think there would be an additional heat flow, but the coefficients should be the same. That is, the heat flow by infiltration, and the heat flow by conductance are separate factors. Both may be going on at the same time. When there is infiltration through the wall, the temperature drop between the air at the two sides is reduced, as indicated by the slide, then, of course, the heat flow by direct transmission is reduced and the coefficients are reduced. But if the temperature drop between the air of the two sides is maintained, I think the heat transmission by straight conductance would be the same and the coefficients should be the same; that is, all other factors being the same, but the heat transmission by infiltration would be increased. In other words, heat is being transferred through the wall by two distinct methods. As to the effect of humidity on the conductivity of a wall, it is evident that, if air passes from the warm to the cold side of the wall, its temperature will be lowered, and if it reaches the dew point, moisture will be deposited in the wall. This will increase the conductivity of the wall.

I might add that Mr. Shodron's emphasis on air leakage brings out an important point in connection with heat transfer and should be carefully considered. Often there is much more heat transmitted by air leakage than by straight conductivity.

PRESIDENT CARRIER: The factors given by Professor Rowley apply to one surface. If you have two surfaces, one inside and the other outside, the total overall conductivity is not of the order shown.

This paper shows that it is of great importance to consider wind velocities on surfaces of high conductance, thin surfaces of any character like glass and it is even important on thin walls.

No. 915

CONDUCTIVITY OF CONCRETE

By F. C. Houghten [1] *(MEMBER)* and Carl Gutberlet [2] *(NON-MEMBER)*
Pittsburgh, Pa.

THE necessary design data for estimating heat loss from buildings have been greatly improved through research during past years. Probably the greatest need for further refinement is for additional facts concerning the variation in conductivity of concrete and stone masonry walls with mixture of ingredients, nature of the aggregate and ageing.

Investigators have reported heat transfer values varying over a wide range and indicating an average coefficient considerably higher than the most widely accepted value. This variation may in some measure result from the different test methods used, but to a greater extent it no doubt reflects the true variation in conductivity of the samples tested. Tamping, or working the wet concrete mortar, eliminates air voids and makes a more dense structure having a higher coefficient of conductivity. Variation in percentage and size of particles of cement, sand and aggregate also affect the number and size of air cells and the density of the mass. The thermal conductivity of concrete depends in a large measure upon the conductivity of the individual particles of sand, gravel or broken rock, which in turn depend upon the geological structure of these materials. The cement chemist has long recognized that important chemical changes take place in concrete for several years after it is poured. These changes affect its chemical composition and density and should logically be accompanied by changes in conductivity.

Results of Various Investigators

Table 1 gives values for conductivity of concrete reported by various investigations and shows values ranging from 2.3 to 17.4 Btu per square foot per hour per degree temperature difference per inch of thickness. The most widely accepted value is that reported by Willard and Lichty as 8.31[a] This value has

[1] Director, A. S. H. V. E. Research Laboratory, Pittsburgh, Pa.
[2] Research Assistant, A. S. H. V. E. Research Laboratory, Pittsburgh, Pa.
[a] See Bibliography.

Presented at the 38th Annual Meeting of the American Society of Heating and Ventilating Engineers, Cleveland, Ohio, January, 1932, by F. C. Houghten.

TABLE 1. CONDUCTIVITY OF CONCRETE ACCORDING TO VARIOUS INVESTIGATORS

Authority	Mix	Age	Thickness Inches	Method of Test	Conductivity k Average Reported	Conductivity k Highest Value	Conductivity k Lowest Value	Mean Temperature F
A.S.H.V.E. GUIDE	1:2:4	....	......	......	8.30	....	....	....
Willard and Lichty	1:2:4	....	3.19	Hot air box	8.31	8.74	7.89	....
Norton[g]	1:2:4	....	......	Electric hot plate	8.42	....	....	2012.
Krueger and Eriksson[h]..	1:5:0	105 Days	4.	Hot air box	9.4	....	....	36.77
" ...	"	105 "	8.	"	9.2	....	....	36.41
" ...	"	105 "	16.	"	9.5	9.6	9.4	37.13
Carmen and Nelson.....	1:1.2:1.1	28 Days	2.5 cylindrical shell	Electric input	9.72	12.48	6.97	....
"	"	120 "	"	"	9.61	11.03	8.70	....
"	1:1.9:1.7	28 "	"	"	11.55	13.93	9.00	....
"	"	120 "	"	"	10.59	12.48	9.57	....
"	1:2.4:2.3	28 "	"	"	10.92	13.64	7.55	....
"	"	120 "	"	"	9.78	10.74	8.13	....
"	1:3.1:3.0	28 "	"	"	9.58	13.06	8.13	....
"	"	120 "	"	"	8.62	10.74	6.10	....
"	1:4.3:4.0	28 "	"	"	11.03	12.77	7.55	....
"	"	120 "	"	"	11.03	10.74	9.29	....
"	1:5.6:5.1	28 "	"	"	9.87	11.03	6.97	....
"	"	120 "	"	"	11.23	12.77	10.16	....
"	"	150 "	"	"	....	17.41	11.03	....
A.S.H.V.E. Lab.	1:3:5	7 Years	26.2	Heat flow meter	11.35	....	....	44.
"	"	10 "	"	"	11.80	....	....	....
"	....	4 "	9.0	"	12.50	....	....	....
"	1:2:5	20 Days	7.94	"	16.26	....	....	80.
"	"	214 "	"	"	14.36	....	....	79.
"	"	3.4 Years	"	"	13.20	....	....	78.
"	1:2:4½	....	8.06	"	14.42	....	....	78.
"	"	142 Days	"	"	13.37	....	....	79.
"	"	3.8 Years	"	"	13.00	....	....	78.
"	....	1 Year	12.	"	7.08[a]	....	....	51.7
A.S.H.V.E. GUIDE	....	....	12.	......	6.22[a]	....	....	....
Van Dusen	....	....	......	......	10.00	....	....	....
Rowley	1:2:4	643 Days	5.95	Guarded hot box	11.90	....	....	39.95
"	1:2:3½	291 "	6.07	"	12.33	....	....	39.99
"	"	303 "	6.04	"	12.66	....	....	39.93

[a] 12 in. hollow concrete block—apparent conductivity including air space.
[g] [h] See Bibliography.

served as the basis for the recommendation of 8.0 and 8.3 for the conductivity of concrete used in tables in past issues of THE A. S. H. V. E. GUIDE.

Carmen and Nelson [b] in 1921 made a long series of tests on forty-eight concrete samples of six different mixtures ranging from 1 : 1.2 : 1.1 to 1 : 5.6 : 5.1. The method of test used by Carmen and Nelson was unique when compared with methods of other investigators. These tests were made on concrete cylinders, 7½ in. in diameter by 24 in. long with an electrical heater placed in a cylindrical hole, 1½ in. in diameter, through the axis of the concrete. The conductivity values were calculated from the electrical energy input and the temperature drop through the cylindrical shell. Tests were made approximately one and four months after the cylinders were poured. Before each test, the sample was heated to a temperature above the boiling point of water to insure perfect dryness. The entire series of tests showed a variation in conductivity from 6.1 to 17.4 which bore no definite relation to either the mix or age of the samples. The average conductivity of all tests was 10.29.

The A. S. H. V. E. Research Laboratory, in making a survey of the heat flow characteristics and conductivity of walls with a Nicholls heat flow meter, reports values of 11.35, 11.8 and 12.5. The values of 11.35 and 11.8 were found for a 26-in. concrete wall in the basement story of the U.S. Bureau of Mines building. The value of 11.35 was reported [c] by Mr. Nicholls in 1923 for a section of the wall several feet above the ground level. The value of 11.8 was obtained by Mr. Zobel on a section of the wall, also several feet above the ground level, in another part of the building in 1926. The value of 12.5 was determined by the Laboratory for a 9-in. basement wall above the ground level in a four year old residence. Dr. Van Dusen of the United States Bureau of Standards reports a value of 10 for a single test on a wall for which there is no record of the mix or age other than that the aggregate was sand.

The A. S. H. V. E. Research Laboratory reports [d] a conductance of 0.59 Btu per square foot per hour per degree temperature difference from surface to surface for a 12-in. hollow concrete block wall. This gives an apparent conductivity of 7.08 Btu per square foot per hour per degree temperature difference per inch thickness for the entire 12-in. thickness including air space and concrete. This wall was in the basement of a residence which was one year old. However, the age of the blocks before laying is not known. THE A. S. H. V. E. GUIDE, 1931, gives the conductance of a similar wall as 6.22 Btu per square foot per hour per degree temperature difference from surface to surface. Prof. F. B. Rowley reports [e] conductivities ranging from 11.34 to 12.66 for three samples of concrete wall tested in a guarded hot box at the University of Minnesota.

The wide variation in conductivity of concrete reported by different investigators points to the desirability of making a survey of the variation in conductivity of concrete and masonry walls due to mix, character of aggregate used, and other factors which may be effective.

It is of interest to note that the value of 8.3 used in THE A. S. H. V. E. GUIDE, 1931, is about the lowest limit of those reported with the exception of a few values obtained for unusual test conditions. All of the data cited indicate

[b c d e] See Bibliography.

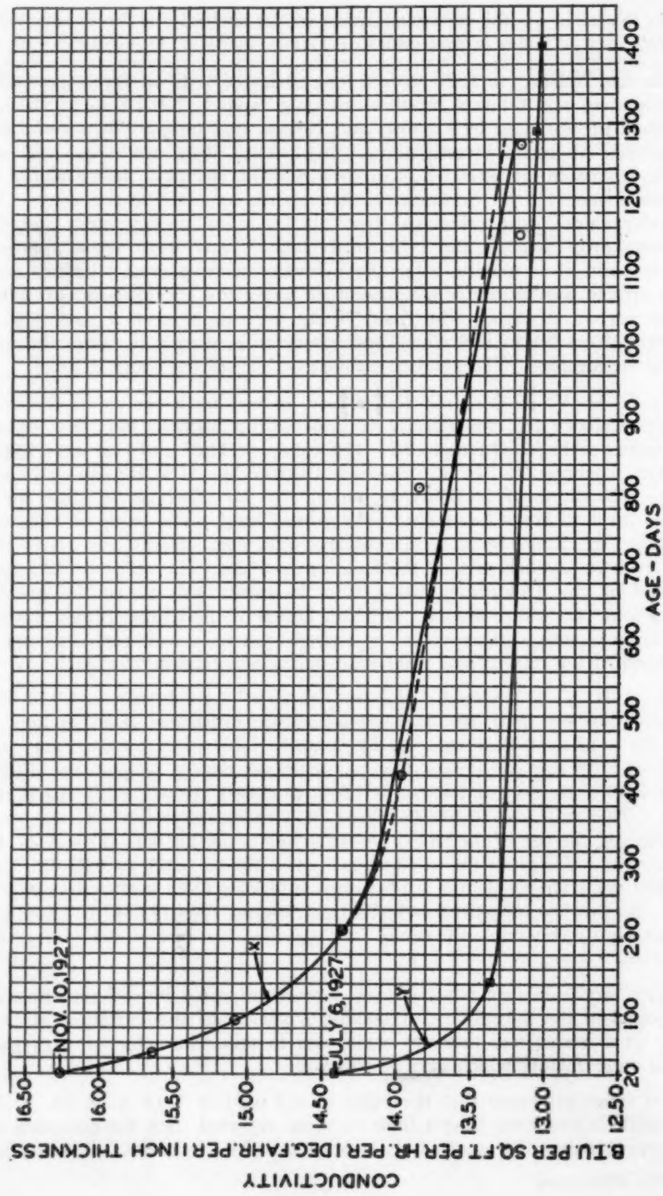


FIG. 1. VARIATION IN CONDUCTIVITY OF CONCRETE WITH AGE

an expectancy for a range in conductivity of concrete walls more than two years old of 8 to 14 and would further indicate that a value of 12 would be more commonly met with in practice than the value of 8.

In order to protect the interests of the designing engineer and contractor, as well as the owner, and to guard against failure, it would seem desirable to accept a value above the average rather than below, until such time as sufficient data may be made available to allow the intelligent use of different values applying to different grades of concrete. A value of 12 is recommended.

VARIATION IN CONDUCTIVITY OF CONCRETE WITH AGING

Failure of well designed heating systems to satisfactorily handle the heating load of concrete buildings during the first and sometimes the second year after completion has led many heating engineers and contractors to the belief that heat loss from concrete buildings is larger during this period than during subsequent years.

This lack of confidence on the part of the engineer and contractor, in the application of the design data for estimating the heat loss from concrete buildings during the first and second year, led to an investigation by the Research Laboratory of the effect of ageing on the conductivity of concrete. A study of two concrete slabs, 8 in. thick by 2 ft square, was undertaken in 1927. Both slabs were well tamped and worked when poured. Slabs *X* and *Y* had cement, sand, gravel ratios of 1:2:5 and 1:2:4½ respectively, and in the spring of 1931 their densities were 137.5 and 139.9 pounds per cubic foot. Both slabs were housed in the A. S. H. V. E. Research Laboratory at Pittsburgh from the time of their pouring where they were subjected to the prevailing indoor atmospheric conditions.

The curves in Fig. 1 show the variation in conductivity of the two slabs with time after pouring. Both show a very definite decrease in conductivity, the greatest rate of decrease taking place during the first two hundred days of the life of the slabs, but apparently continuing for more than three years, or until the day of their last test, when the samples were 1275 and 1405 days old respectively. The conductivity values are for mean temperatures between the warm and cold surfaces of the slab ranging from 77 to 80 F.

The different points show a variation from the mean curve as drawn, ranging to a maximum of 0.2 Btu or ±1.5 per cent. This percentage is about double the percentage of error which would be anticipated due to the method of test. These errors must be accounted for, in part, as irregular variations in conductivity of the slab superimposed upon the progressive change shown by the curve, and may result from change in moisture content or other factors.

The two samples of concrete give curves showing different slopes or rate of change in conductivity with time, particularly during the last two years. There is no apparent reason for this difference. Sample *Y* was not tested as often as sample *X*, and little is known of its conductivity from 142 to 1289 days after pouring. A knowledge of changes taking place during this time might result in a curve of somewhat different shape.

The solid line curve best fitting the conductivity points for sample *X* shows an appreciable rate of change in conductivity for the present time, which if continued during subsequent years, would indicate a material effect on heat

loss from concrete buildings throughout their life. However, it should be pointed out that the apparent irregular variation in conductivity from time to time, as indicated by the failure of the points to fit the curve more closely, may account for a considerable part of the apparent slope of the curve as drawn for the past two years. The dotted line curve for sample *X* shows another possible way of drawing the curve through the points with no greater maximum distance from any points to the curve than shown for the solid line curve. It would also predict a very different future for the conductivity of

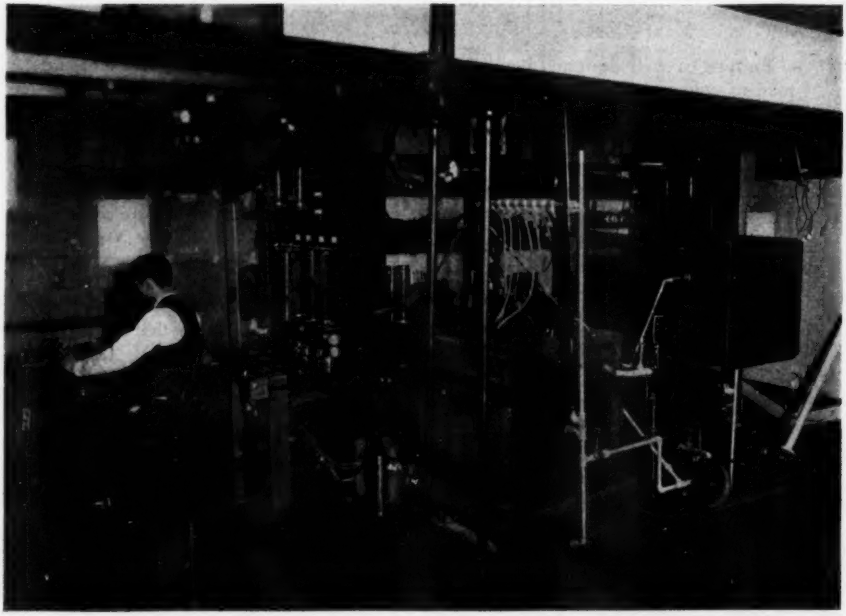


FIG. 2. APPARATUS FOR DETERMINING THERMAL CONDUCTIVITY

this sample. The difficulty in establishing the curve and its prediction for the future from the few points available indicates the desirability of occasional tests on these samples during the next several years, or until the change in conductivity with time can be shown to have ceased.

During the first year after pouring, the respective samples *X* and *Y* showed a decrease in conductivity of approximately 13 per cent and 8 per cent. From the end of the first year to the time of the last test, the decrease amounted to approximately 6 per cent and 2 per cent, or an average yearly rate of decrease after the first year of 2.5 and 0.7 per cent. The total percentage decrease in conductivity since pouring is 18.8 and 9.8 per cent for the respective samples.

The higher conductivity of concrete indicated for the first and second years after pouring would result in a very noticeable increase in heating load during

the first season, and some increase the second season, over subsequent years for a building whose walls consist largely of concrete without insulation or finishing on the inside. It should be noted, however, that for walls consisting of concrete and some other material, having a considerable higher resistance to the flow of heat, the effect of variation in conductivity of concrete with age would have a smaller percentage effect on the heating requirement for the first year.

It should be noted that the conductivity shown by these two samples is higher

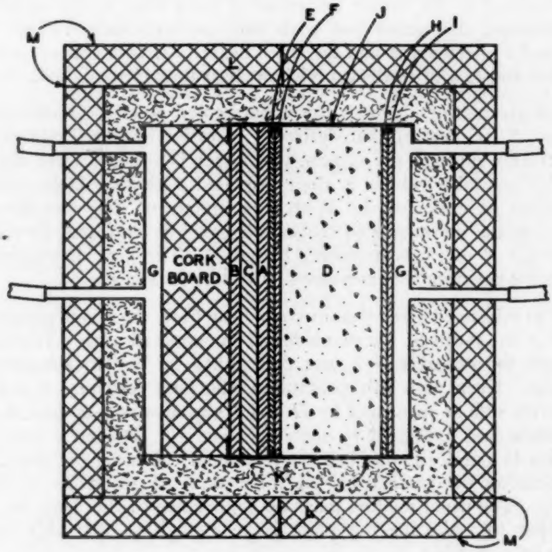


FIG. 3. GUARDED HOT PLATE FOR DETERMINING THERMAL CONDUCTIVITY

A—Main hot plate.
B—Auxiliary hot plate.
C—High resistance heat meter.
D—Concrete test sample.
E & F—Heat meters, warm side.
G—Water cooled heat absorber.
H & I—Heat meters, cool side.
J—Wax paper.
K—2 in. hair felt.
L—2 in. corkboard.
M—Waxed surface and joints.

than any other value listed in Table 1 with the exception of certain tests by Carmen and Nelson. The only apparent reason for this is that these samples were probably worked and tamped more during pouring than is the usual practice.

METHOD OF TESTS TO DETERMINE VARIATION OF CONDUCTIVITY OF CONCRETE WITH AGE

Fig. 2 is a photograph and Fig. 3 a drawing of the test apparatus. The samples were tested in the double guarded hot plate designed and built at the

A. S. H. V. E. Laboratory by P. Nicholls.[f] *A* is a 2-ft square, main guarded hot plate. *B* is an auxiliary guarded hot plate, separated from *A* by a high-resistance, heat-flow meter *C* approximately ½ in. in thickness. In operating the double guarded hot plate, the electric current through the center and also that through the guard ring of heater, *A*, are adjusted to allow no temperature difference between the center and guard ring. Heater *B* is likewise adjusted to give no temperature difference between its center and guard and at the same time no heat flow through *C*. With this condition prevailing, it is obvious that all heat generated in the center heater of *A* must flow in the direction of *D*. *D* is the concrete slab under test with two low-resistance Nicholls heat flow meters, *E* and *F*, between it and the main heater. *G* is a copper, water-cooled, heat absorber and *H* and *I* are two Nicholls heat flow meters similar to *E* and *F*.

The water circulating through the water-cooled plate, *G*, is thermostatically controlled to ±0.05 deg of that desired. In order to guard against heat loss by air infiltration through the system, wax paper is sealed around the perimeter of the entire assembly which is then insulated with two inches of hair felt and two inches of cork board. With this equipment, the rate at which heat was supplied to the warm side of the sample could be measured by the electrical input of heater *A* or by heat meters *E* and *F*. The heat dissipated from the cold side of the concrete was measured by meters *H* and *I*.

In order to minimize error due to stray heat flow, the room temperature was maintained at the mean of the two sides of the concrete slab. The temperature drop through the concrete slab was determined by five thermocouples placed on each face. Even with this precaution, the heat recovery at the cold side of the concrete slab as measured by *H* and *I* was always less than that supplied at the hot side and measured by meters *E* and *F*. This is a fact worthy of consideration in the designing and application of the usual hot plate method to the determination of conductivity of thick slabs of materials having a high conductivity. The usual method takes no cognizance of this loss, the customary practice being to make the hot plate large enough so that this loss may be assumed to be negligible.

In calculating the conductivity, the average heat flow indicated by the meters on the warm and cold sides and the temperature drop through the concrete slab were used in the usual formula:

$$k = \frac{qL}{A(t_1 - t_2)}$$

where

k = thermal conductivity (heat transferred in Btu per square foot, per hour, per degree difference F per inch of thickness).

q = heat transferred per unit time (Btu per hour).

L = thickness of path of heat flow (inches).

A = Area (square feet).

t_1 = temperature, warm side (F).

t_2 = temperature, cold side (F).

---

[f] See Bibliography.

Summary and Conclusions

This report points out some of the reasons for wide variation in conductivity of concrete and indicates that the value commonly used is probably too low an average to properly protect the practice of the consulting engineer and contractor. An average value of 12 is recommended until more is known concerning the effects concrete characteristics have on the conductivity. The decrease in conductivity of two slabs of concrete with age is shown by curves to be of considerable magnitude and progresses at a decreasing rate for at least 30 months after pouring.

Bibliography

[a] University of Illinois Engineering Experiment Station Bulletin No. 102.
[b] University of Illinois Engineering Experiment Station Bulletin No. 122.
[c] Practical Applications of the Heat Flow Meter, by Percy Nicholls (A. S. H. V. E. Transactions, Vol. 30, 1924).
[d] Additional Coefficients of Heat Transfer as Measured Under Natural Weather Conditions, by F. C. Houghten, Carl Gutberlet and C. G. F. Zobel (A. S. H. V. E. Transactions, Vol. 35, 1929).
[e] To be published later.
[f] Measuring Heat Transmission in Building Structures and A Heat Transmission Meter, by Percy Nicholls (A. S. H. V. E. Transactions, Vol. 30, 1924).
[g] Some Thermal Properties of Concrete, by Charles L. Norton (A. S. M. E. Journal, 1913).
[h] The Academy of Engineering Science, by H. Krueger and A. Eriksson (NR7 Stockholm, 1924).

DISCUSSION

F. B. Rowley (Written): The authors have brought out the important fact concerning the thermal conductivity of concrete, that there is a variation depending upon the age of the concrete and probably upon the character of the aggregate, proportion of mix, etc.

Three 6-in. concrete monolithic walls were built and tested by the hot-box method at the University of Minnesota with the results shown in Table B.

Table B

Wall No.	Age of Wall	Thermal Conductivity k
55	1 month, 5 days	12.3
55	4 months, 6 days	11.2
55	11 months, 15 days	11.6
55	22 months, 3 days	11.8
55	36 months	10.7
68	10 months	12.4
69	9 months, 17 days	12.1

Wall No. 55 was built with a 1:2:4 mix and Walls No. 68 and 69 were made of a 1:2½:4 mix. Wall No. 68 was mixed with water to give a 6-in. slump for the mixture, and Wall No. 69 was mixed with water to give a 3-in. slump for the mixture. In each wall, No. 4 sand was used; that is sand of which 95 per cent passed a No. 4 sieve. The gravel used was graded from 1½ to ¼ in. diameter.

While these tests did not cover a sufficient range in concrete mixtures to be conclusive, the results do substantiate the conclusions of the authors in that a higher coefficient should be used in making calculations for concrete structures. The tests

on Walls No. 68 and 69 have not as yet been completed, but those made on Wall No. 55 show a reduction in conductivity after the first 4 months. For the next 18-months' period, there was no reduction shown, although for the 3-year period there was a reduction to 10.7. It is possible that some of this irregularity may be accounted for by the different amounts of moisture in the concrete as absorbed from the atmosphere.

EDGAR C. RACK (WRITTEN): I am in general accord with the conclusions of the authors after their extended survey and analysis of this problem. The only questions are whether the value of 12 should apply as a general conductivity factor for all concrete, and whether this value, without qualifications, may not be somewhat more than conservative for the use recommended.

Concrete, in general, necessarily shows more than ordinary variations in its kind and composition than do most other materials commonly used in the building industry. It may be said that the individual constituents (including the manufactured light weight aggregates which are coming into more or less common use) going into the mix of various grades of concrete may all be considered of nearly the same specific gravity. Therefore, as is generally considered, they should have approximately the same thermal conductivity at the same density. Variations in densities of the constituents and the resulting concretes are primarily due to the size of the enclosed pores and their count per unit volume. It would then seem logical, as has been suggested, that some definite relation be established between the density of the concrete and its thermal conductivity. The recommended value of 12 could, on this account, be qualified to apply for materials falling between certain limits of density.

The thermal conductivity of concrete, expressed as a function of the mean temperature, increases only slightly with increasing mean temperatures. However, it might have been well if the conditions of test selected by the authors had been such as to allow the thermal conductivity to be expressed at a mean temperature more nearly that commonly used as a basis for calculations of heat losses through building constructions.

It has been found, in general, that tests made on thin sections of air-dried concrete give lower conductivity values than those resulting from tests made on thicker specimens of the same material. However, when the test slabs are heated to a nearly bone-dry condition and retested, the conductivity values for both thick and thin sections agree very closely. It is felt that the decision to test relatively thick slabs, and allow them to come only to an air-dried condition before test, gives a more representative value of conductivity for concrete than if the usual procedure of oven-drying the samples before test had been followed. It might be safely assumed that in all building walls the concrete will be thoroughly dry only in a relatively thin section of that portion of the walls enclosing the heated air spaces of the building.

It is believed that a study of additional data not presented in Table 1 is justified While the authors did not so state in their report, they may have included in so far as possible in their survey and analysis of the test results of others, the factors of density, moisture content, and the water-cement ratio of the mixtures for the various specimens.

With reference to the test method used, the authors have noted their inability to obtain 100 per cent heat recovery from the cold face of the specimen. In our laboratory, and especially when true temperature readings and reliability of results are desired, our experience has been such as to indicate a minimum of 1.0 as the ratio of the guard ring width to the thickness of the test specimen. A minimum value of 80 to 1 is also taken as the ratio of the area of the sample to its thickness. Insufficient data are given in the report under discussion to determine what these ratios were for the particular apparatus used. It is believed that data of this nature should be included in such a report for the benefit of those making future reference to this important piece of work.

It has further been our experience with specimens which have hard or slightly uneven surfaces that more than ordinary care is essential in attaching the thermocouples. The method used for their attachment to the face of the specimen should also be given careful consideration. These precautions are necessary in order to insure good thermal contact and more nearly true readings of surface temperatures. With the materials under investigation showing relatively high thermal conductivities, and with comparatively small temperature difference maintained between the hot and cold faces, small variations from the true temperature gradient through the specimen will result in significant errors for calculated thermal coefficients.

Any change such as recommended, when adopted, will necessitate recalculations for and revisions of a considerable part of the data appearing in Chapter 3 of THE GUIDE 1931.

Concrete is one of the more important and most commonly used materials in the building industry. It is also one for which we probably have the least positive coefficient of thermal conductivity to offer for use by the engineer and architect in making heat loss calculations. The authors and compilers of this report are to be complimented for their work in a field of endeavor which is in much need of clarification and in which more reliable information is to be welcomed.

E. R. QUEER (WRITTEN): In 1918 a slab of concrete 3 ft x 3 ft and 4$\frac{1}{16}$ in. thick was poured at the Engineering Experiment Station at Pennsylvania State College for the purpose of making thermal conductivity tests. A fire destroyed all the early data

TABLE C

Mix	Age (years)	Thickness Inches	Method of Test	Conductivity	Mean Temp.
			SERIES I		
Unknown	7	4$\frac{1}{16}$	Heat-Flow Meter	12.16	69.1
				11.76	68.7
				12.09	68.7
				11.78	72.6
				11.84	73.1
				12.52	77.8
				11.92	73.9
			SERIES II		
Unknown	9	4$\frac{1}{16}$	Heat-Flow Meter	12.35	83.4
			Heat-Flow Meter	12.45	206.0
			Electrical-Input	13.10	83.4
			Electrical-Input	13.10	206.0
			SERIES III		
Unknown	11	4$\frac{1}{16}$	Electrical-Input	11.2	48.5
				11.1	59.2
				11.5	73.2
			SERIES IV		
1-2-3	2	1½	Electrical-Input	8.65	87.3

and the mix is unknown. A rather coarse aggregate (crushed rock) was used giving a rough surface.

Table C shows the available record of tests conducted on this concrete.

The slab had been stored in a shed, where the humidity was not high, until 1925, when the first series of tests were made. In this series the slab was placed next to an ice tank with a heat-flow meter separating the tank and the slab; and a heat-flow meter on the surface facing the room. Heat was supplied to the air in the room by shielded electric heaters. The heat input was quite low giving a small temperature difference. The average value of k for 7 separate tests was 12.01 at a mean temperature of 72 F. Both heat-flow meters indicated about the same amount of heat flowing through the specimen.

Between the first and second series of tests the slab was stored in a room where the relative humidity remained at 90 per cent. In test Series II, the heat input to the test blank was measured both electrically and with heat-flow meters. This is a duplicate of the method used at the A. S. H. & V. E. Research Laboratory. The conductivity shows an increase over the earlier tests but the value at 206F-MT should apparently be higher, unless the moisture was being driven off at this high temperature. There is an apparent discrepancy between the measured electrical heat input and that indicated by the heat-flow meters. The meter facing the ice tank always read lower than the electrical heat input. The authors have also mentioned this discrepancy, and no plausible explanation is available. Although a larger surface area and a thinner blank was used in the tests at the Engineering Experiment Station, the same result was evident.

During the period between Series II and Series III the slab was subjected to drying. In the latter series the heat-flow meters and ice were not used. Heat from the slab was dissipated directly into a constant temperature cold room.

The value of 12 appears to be a fair average for the conductivity of stone aggregate concrete.

In Series IV a 12-in. x 12-in. blank 1½ in. thick was used. The specimen was well dried.

A comprehensive series of tests was made[3] on cinder concrete blocks. The conductivity in a Btu per square foot per hour per degree Fahrenheit per inch of a 4-in. solid block wall was found to vary from 4.2 to 4.8 at 70 F mean temperature, depending on the type of cinder used. The conductance of a 12-in. wall was found to be 0.46 Btu per square foot per hour per degree Fahrenheit with the wall sealed at the top. Such things as number and arrangement of air spaces, closing the flues, no mortar on middle web, plastering and stuccoing, and varying the type of cinder used cause marked changes in the conductance of the wall. One of the most annoying factors was that plastering and stuccoing a wall increased the conductance rather than decreasing it, as would be expected.

F. R. McMILLAN[4] (WRITTEN): While we have not carried out any studies on heat conductivity at normal temperature, our general investigations of the properties and uses of concrete have developed information that prompts some comments on the data in this paper.

That wide differences in conductivity of concrete are obtained by different investigators is not at all surprising, because there is a wide range in the physical structure of different concretes and, what is probably of greater importance in the heat conductivity, a wide range in possible moisture content.

It is probably the difference in moisture content that accounts for the large difference in conductivity at early ages shown for the two specimens in Fig. 1. The significance of this will be more readily understood from a simple analysis of the makeup of a concrete mixture. For the purpose of illustration I have given in Table D comparative data for two mixes of the proportions indicated having approximately the weights per cubic foot reported for samples X and Y of the paper. Necessarily, these are based on certain assumptions regarding the aggregate, consistency of the concrete, etc. As to the essential fact which it is desired to bring out, however, these data can be taken as representative of conditions closely approximating those which obtained in the tests. The figures in the table are for 1 cu ft of concrete.

The point it is desired to bring out by Table D is that more than 80 per cent of the water used in mixing will ultimately be driven off under the conditions of storage and testing to which these specimens were subjected. At the time of the last tests, shown in Fig. 1 of the paper, they do not differ greatly in conductivity which is to

[3] See Bulletin No. 37—*Heat-Flow Meters and Thermal Conductivity Measurements*, by F. G. Hechler and E. R. Queer, Engineering Experiment Station, Pennsylvania State College.
[4] Director of Research, Portland Cement Assn.

Table D

	Weights of Ingredients—lb.	
	1-2-4½ Mix	1-2-5 Mix
Cement	17.1	15.8
Water	10.5	11.0
Aggregate	121.0	120.0
Total = weight per cu ft when mixed	148.6	146.8
Water lost by evaporation and drying	8.5	8.9
Dry weight per cu ft	140.1	137.9
Voids by volume ot concrete	13.6%	14.2%

be expected on the basis of the dry weights and percentages of voids indicated in the table.

The significant feature of the comparative conductivities of these two concretes is that in the tests at the early ages. It appears that the first tests for both specimens were started when they were 20 days old, but as the specimens were made at different times, the conditions of curing during the first 20 days could differ widely. It would not be inconsistent with the known facts regarding concrete, for the amount of water lost in the first 20 days to vary from a small percentage to as much as 35 or even 50 per cent of the total quantity used in mixing. This being the case, it is easy to understand the wide difference indicated in the early tests.

Another factor that deserves consideration in a study of these results is the temperatures at which the tests were carried out. From the foregoing comments it can be seen that the amount of water retained in a concrete mixture is largely a function of the drying condition to which it has been exposed. It is not stated in the report just what temperatures were maintained at the two faces of a specimen during test. It can be seen that in a material like concrete the temperatures used would be a matter of considerable importance. For example, if the tests on the two sides were maintained at 40 and 120 F, drying conditions would be somewhat more severe than for temperatures of 0 to 80 F. Under the latter condition, the conductivity shown would be somewhat greater, but probably more nearly in accord with the conditions existing in ordinary building practice. If the tests at the respective periods had all been carried out at temperatures more nearly approaching those to be encountered in an ordinary building, it is certain that the amount of water lost would have been considerably less and the conductivity shown would have been considerably greater.

In the foregoing, nothing has been said about the effect on the results of the tests of the latent heat of vaporization of the quantity of moisture driven off, while the test is in progress. It can be seen that under certain conditions of test, this factor might be of considerable importance and materially affect the actual test results.

Another factor not previously considered is the effect of the type of aggregate on the heat conductivity. In an extensive series of tests carried out in the Research Laboratory of the *Portland Cement Association,* covering the performance of concrete masonry walls when exposed to standard fire test conditions, some useful information on this factor has been obtained. These tests have shown that the thermal conductivity of concrete at high temperatures can be varied between rather wide limits by changes in the character of the aggregate. While the conditions of these fire tests are not at all comparable with those obtaining in tests such as reported in the paper, there is every reason to believe that similar variations in the conductivity under normal temperatures will obtain. For a complete report of these tests, see paper by C. A. Menzel on Tests of the Fire Resistance and Stability of Walls of Concrete Masonry Units in the Proceedings of the *American Society for Testing Materials,* V. 31, Part 2, page 607, 1931.

F. C. HOUGHTEN (WRITTEN): As pointed out by Mr. McMillan, there are a number of variable factors effecting the conductivity of concrete, although understanding of the conductivity of concrete cannot be had without an extended investigation of each of these independent factors. The object of the Laboratory in making this study was not to investigate the effect of each of these variables but rather to, *first,* furnish data on the controversial question raised by heating engineers and contractors as to whether or not a concrete building had a greater heating demand during the first few heating seasons than in the subsequent years and to establish the approximate increased demand during the aging period; *second,* to investigate the accuracy of the generally accepted conductivity of concrete as used in estimating the heating load and if it were found to be in error to establish a value for such conductivity which would insure the installation of sufficient heating capacity to protect the designing engineer and heating contractor.

While it is recognized that an extended investigation of the variable factors effecting the conductivity of concrete is desirable, it is questionable if the cost of such an investigation should be borne by the heating engineer alone. A greater cooperation on the part of other groups interested in the use of concrete in building constructions would seem desirable.

A. B. SHENK [5] (WRITTEN): My present interest in the conductivity of concrete produced from Haydite aggregate, is the result of studies made for the Federal-American Cement Tile Co.

Prior to the time that the company adopted the use of Haydite as an aggregate and was making roof slabs of the ordinary type of concrete, we used in our computations a thermal coefficient of 8.3 Btu per hour, per square foot, per degree Fahrenheit for concrete. Computations using this figure were in various instances used in connection with heating as well as in connection with the elimination of condensation. The performance after the buildings were completed indicates that the computations based on the foregoing coefficient were conservative. The same thing holds true of similar computations on Haydite concrete, when using a much lower factor of thermal conductivity.

It seems to me that since a value of thermal conductivity of from 6.0 to 8.3 for ordinary concrete has been used for a number of years in designing the heating equipment on numerous buildings all over the country with apparent satisfaction, there is considerable argument in favor of continuing the use of such coefficients.

The American Society of Refrigerating Engineers in the report of its Insulation Committee gives a number of tables showing the heat transmission of insulating materials, and include therein numerous building materials including concrete. Examination of these data fail to reveal any coefficient for ordinary concrete higher than 9.04, and most of them are considerably lower. It is only reasonable to assume that these coefficients have been used in many cases in designing the refrigeration necessary in reinforced concrete buildings, constructed for cold storage purposes, and we want to believe, in light of no definite evidence to the contrary, that these buildings have performed their purpose satisfactorily. This again substantiates the fact that the present coefficients of conductivity of concrete give satisfactory performance when used in designing structures involving heat loss.

In view of the foregoing, I cannot help but feel that the Society should move slowly and consider carefully before recommending the adoption of higher coefficients of conductivity for ordinary concrete, thereby penalizing unnecessarily this excellent structural material. Of course, it goes without saying that the various coefficients previously mentioned are considered as applicable only to concrete produced using the usual sand and gravel aggregate, and do not apply in any way to concrete produced from burned shale cellular aggregate.

---

[5] Engineer, Federal American Cement Tile Co.

J. H. BRACKEN (WRITTEN): I think this paper, although it is specifically directed to concrete, reveals a general danger to which the heating engineer is exposed and that is that ordinary building materials are complex in their qualities and not simple as we are apt to think.

This paper shows the wide variation in conductivity of concrete, as found by various investigators, and which is due to variations in density, porosity, and age. It is likely that in any concrete wall for example there is a wide variation between one part of the area and another part of the area, and perhaps no two parts of the area are identical in density or porosity.

I wish to urge, therefore, that the Society adopt the recommendations of this paper and use a value of 12 as the conductivity of concrete. I think if there is any fault at all to be found in this conclusion it is because of its conservatism. Personally I should like to see this conductivity at least 25 per cent higher, but at least the recommended conductivity is a large step in the direction of safety over the conductivity which we are now using.

L. A. HARDING: I think the difference between the lower and the higher values in the conductivity of any material is largely due to the method of testing. These low values of concrete reported for a number of years have, I believe, been invariably made on large size specimens by the hot-box method. The Nicholl's heat meter has always given high values. There is a variation in concrete, of course, as with any manufactured material.

H. P. REID: I have followed with considerable interest Director Houghten's work in Pittsburgh. Some of the discussion points out the necessity of a much wider range in this study before we establish any factors of heat flow or heat losses. Concrete can be designed to have almost any type of heat loss desired, provided that is the objective. Concrete that has a very good insulating value can be made by using certain types of expanded concrete or expanded aggregates.

A study of the work done throughout the United States and in Continental Europe will point to the fact that the conductivity is a factor of the type of aggregate. It makes a difference whether you use limestone, dolomite, pebble, blast furnace slag (as is now being accepted as an aggregate in a great many building and construction projects), or some of the lighter types of aggregates.

There are marked differences also in the conductivities of the concretes when they are made up of varying ratios of the volume of cement to the volume of the aggregate.

The values quoted from Penn State, varying from 8 + to 12 +, without any reason thus far being found to account for it, probably are not due alone to density of the concretes.

Another possible factor to be considered in recent concretes is the finer ground and higher strength cements, made according to new specifications, which are being used in present day concretes. Those who follow the industry know that about two years ago the *American Society for Testing Materials* and the *U. S. Bureau of Standards* cooperating raised the specification on cement which in many cases requires, in order to meet specification, much finer grinding. This finer grinding of the cement results in greater percentages of the infinitely small particles so that a different reaction and different crystalline structure in the concrete may result. This again may affect the conductivity.

These tests are interesting and no one should feel that their accuracy should be challenged, yet some of us do feel that there should be a much wider study made as to the effect of the foregoing possibilities, as well as some others, on the heat conductivity before radical changes are made in the values the Society recommends for general use.

W. L. FLEISHER: There has been no discussion about the fact that the conductivity of concrete varies over a period of two years. This is important because the public expects the heating plant to operate within a definite period after they occupy a building and the practical significance of this thing does not seem to have been brought out. There is no divergence of opinion as to the fact that it does vary materially over a long period and a warning should go out to the engineer and the owner of the variation in his heating requirements over that period. I suggest that this fact should be definitely brought out by the Society.

PRESIDENT CARRIER: The moisture removal from concrete we know requires two years, and during this period buildings have been humidified perfectly without humidifying equipment simply by the evaporation from the concrete or plaster. This necessarily adds a great deal to the heating load, aside from the increase in conductivity, so a big reserve is required in the early stages, as Mr. Fleisher pointed out.

PROFESSOR KNUDSEN: It may be of interest to observe that the sound absorption coefficients of concrete vary in such a manner as would be consistent with this variation with heat conductivity. A large reverberant concrete chamber, shortly after it was constructed, had a time of reverberation at 1024 cycles per second for 12 sec. That is, sound of this pitch would remain audible for 12 sec. After the concrete had aged for a year, this time had reduced to approximately 10 sec. and after it had aged for 2 years it dropped to 8.5 sec. This means that the concrete becomes more and more porous as it ages.

Aging of concrete, therefore, not only improves the concrete from the standpoint of heat insulation, but also from the standpoint of sound absorption. The increase in absorption is especially characteristic of high-frequency sound, which means that the pores in the concrete are very small. The increase in porosity of aging concrete would be expected to produce an increase in both sound-absorption and heat-insulation.

M. C. W. TOMLINSON: One thought that has not been brought out very emphatically in connection with this paper on concrete is the effect of humidity and possibly also the affect of flow of moisture through concrete due to differences in humidity. I think that we are very much in need of some research work on flow of moisture through building materials.

No. 916

TRANSMISSION OF RADIANT ENERGY THROUGH GLASS

By R. A. Miller[1] (*MEMBER*) and L. V. Black[2] (*NON-MEMBER*),
Pittsburgh, Pa.

THE object of this paper is to point out certain facts in connection with the transmission of radiant energy through glass, and if possible, to correct the somewhat prevalent idea that large radiation losses occur through the glass. It should be borne in mind that there are two different forms of energy, namely:

1. Radiant energy, which is an electro-magnetic wave motion.
2. Sensible heat, which is molecular kinetic energy in matter.

Radiant energy is transformed into sensible heat only when it strikes a body of matter and is absorbed. In considering sensible heat, the authors will deal with the absorption of radiant energy by bodies in the room or building, including the atmosphere.

Radiant energy covers a very wide range of waves, from the extremely short waves of the *cosmic ray* to the extremely long waves of radio broadcasting. This discussion is confined to the rather narrow band of radiant energy, included between the wave lengths of 0.292 microns (μ), the limit of solar radiation in the ultra-violet to 30.0 microns, the maximum wave length measured for radiation from a body at 126 F. In this connection it is especially worthy of note that the limit of solar radiation in the infra red reaching the earth is given at a wave length of 5.3 microns.

Emission of Radiant Energy

All the bodies at any temperature above absolute zero are continually emitting radiant energy. Most of the energy radiated from bodies at low temperature, however, is at long wave lengths, and therefore lies in the infra-red portion of the spectrum. If the temperature of such a body were slowly raised the wave lengths of the maximum energy given off would shift toward the shorter

[1] Technical Sales Engineer, Pittsburgh Plate Glass Company.
[2] Research Laboratories, Pittsburgh Plate Glass Company.

Presented at the 38th Annual Meeting of the American Society of Heating and Ventilating Engineers, Cleveland, Ohio, January, 1932, by R. A. Miller.

wave lengths until at about 900 F a considerable portion of this energy would be at wave lengths less than 0.77 microns and would therefore be visible to the eye, and the body would have a dull red color. If heated to a still higher temperature, the wave lengths of the maximum energy would become shorter and shorter and the color of the body would change to yellow and finally to

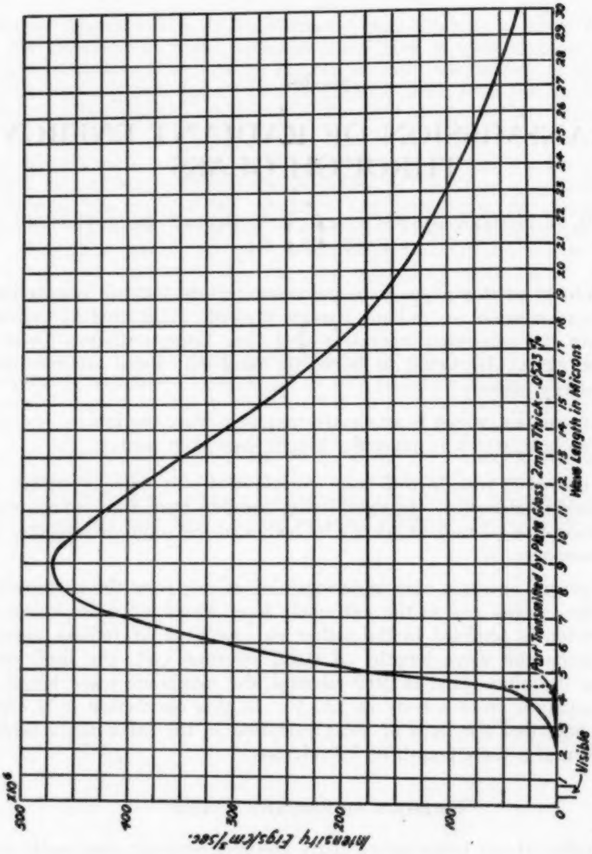


FIG. 1. CURVE OF INTENSITY OF RADIATION AT DIFFERENT WAVE LENGTHS FROM A BLACK BODY AT 126 F

white. At very high temperatures a considerable portion of the energy is at wave lengths shorter than the visible (below 0.38 microns) and would therefore be in the ultra-violet portion of the spectrum.

While glass is transparent to energy in the visible portion of the spectrum, it is opaque to all other wave lengths except a narrow band of the longest of the ultra-violet and shortest of the infra-red. The portion of the ultra-violet

and infra-red that is transmitted to some extent is cut off by thick glass, and even comparatively thin glass cuts off a large part of it.

The amount of total energy radiated at all wave lengths by a body can be represented by the area under a curve plotted with the intensity of the radiation at different wave lengths as ordinates and the wave lengths as abscissae. The

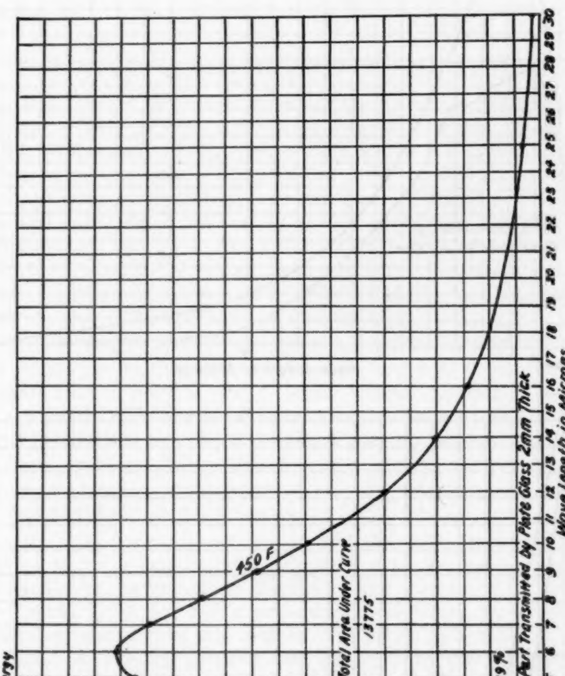
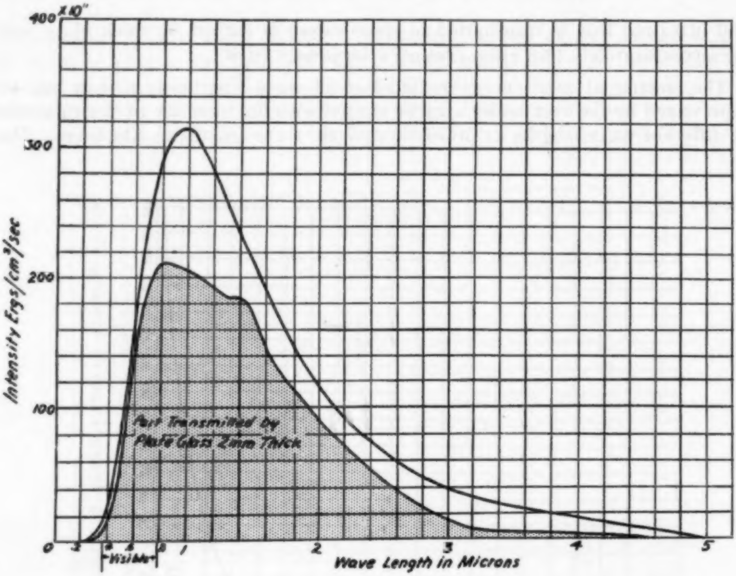


FIG. 2. CURVE OF INTENSITY OF RADIATION AT DIFFERENT WAVE LENGTHS FROM A BLACK BODY AT 450 F

fractional part emitted between any two wave length limits can be determined by comparing the area under this portion of the curve with the total area under the curve. Figs. 1, 2, 3 and 4 were drawn in this manner using the value for intensity at different wave lengths given in the International Critical Tables Vol. V.



FIGS. 3 AND 4. CURVES OF INTENSITY OF RADIATION AT DIFFERENT WAVE LENGTHS FROM A BLACK BODY AT 3000 K* (TUNGSTEN FILAMENT TEMPERATURE) (FIG. 3, TOP) AND FROM A BLACK BODY AT 5000 K* (SUN TEMPERATURE) (FIG. 4, BOTTOM)

* Degrees Kelvin.

The work of Coblentz[3] has shown that ordinary window glass 2 mm thick is opaque to all radiant energy having wave lengths shorter than about 0.3 micron and longer than 4.5 microns. Glass then is transparent to some extent to the ultra-violet between 0.3 and 0.37 microns, the visible portion of the spectrum from 0.37 to 0.78 microns, and the infra-red from 0.78 to 4.5 microns. The amount transmitted, however, is not uniform for all these wave lengths, but is selective for different wave lengths and never reaches 100 per cent in any case.

TRANSMISSION OF RADIANT ENERGY AT VARIOUS WAVE LENGTHS

Fig. 1 is derived from data covering the radiation from a body at 126 F, and from a study of this curve it is evident that the percentage of energy radiated from a body of that temperature which lies between 0.3 microns and 4.5 microns is very small, being in the order of 0.556 per cent of the total energy available. Considering only that portion from 0.3 to 4.5 microns wave length and particularly the heavily shaded area under the curve (Fig. 1), it becomes evident that glass actually transmits only a very small portion of the conceivably transmissible energy, namely, about 9.46 per cent. This percentage becomes insignificantly small when referred to the total radiated energy, and amounts to only 0.0523 per cent of that total.

As already stated, the wave lengths radiated from a body become progressively shorter as the temperature increases, and it is evident from Fig. 2 that radiations from a body at 450 F which conceivably might be transmitted by glass are considerably greater than in the case of a body at 126 F. In this case the energy which is actually transmitted amounts to only 1.19 per cent of the total energy radiated. Figs. 3 and 4 represent the total energy available and transmissible by glass at the temperatures of a tungsten filament in a projection lamp, and of the sun itself, respectively.

This series of curves illustrates the fact that glass does not transmit radiant energy of all wave lengths proportionately, but only that very narrow band of wave lengths in the immediate vicinity of one micron on the chart of relative wave lengths (Fig. 5); and even within these limits, the transmission varies from zero to a maximum and back to zero again, in somewhat irregular order. In no case does glass transmit 100 per cent of the incident energy, since approximately 4½ per cent of the incident visible light is reflected from each surface, and in the invisible portion of the spectrum the reflectance runs as high as 19.2 per cent in the infra-red region.

It is this inability of glass to transmit appreciable amounts of long wave radiation which makes possible the growing of plants in green houses and hot frames. The short waves of solar infra-red radiation pass freely through the glass, are absorbed within the soil and other content, and again irradiated in the form of low temperature rays of long wave length, which are trapped within the hot house.

VERIFICATION OF RESULTS BY TEST

To test the validity of the foregoing calculated values, an apparatus was set up as shown in Fig. 6. An inverted electrically heated radiating surface *A*

[3] U. S. Bureau of Standards Papers, Vol. 14, 1918-19.

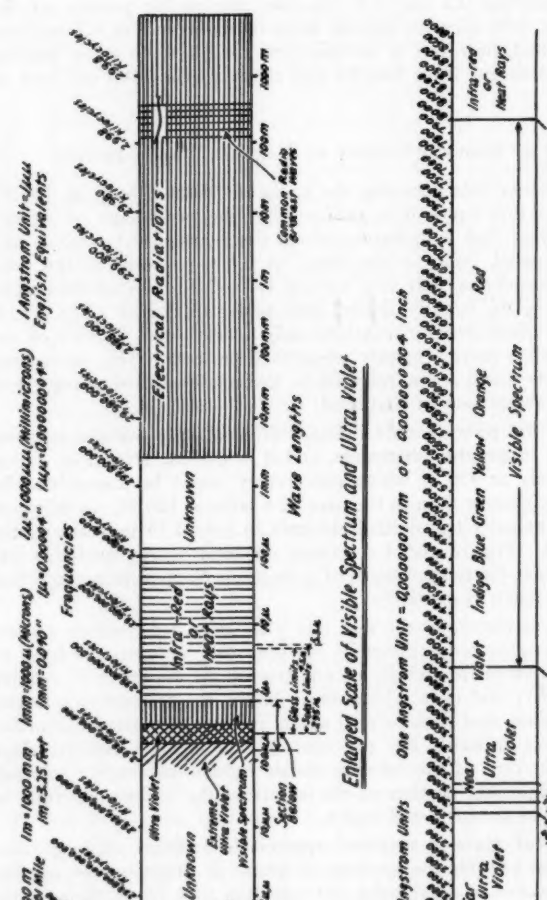


FIG. 5. RELATIVE WAVE LENGTHS OF DIFFERENT FORMS OF RADIANT ENERGY AND CORRESPONDING FREQUENCIES AT 299,800 KILOMETERS PER SECOND (KMPS)

was kept at a definite temperature (450 F) by the automatic control *B*. This temperature was chosen because of the availability of instruments to measure radiant energy emitted at this temperature. The hot inverted surface *A* was protected from drafts by a guard *C*. *D* was a fixed-focus, total-radiation pyrometer specially designed to measure low temperatures. It was trained on the portion of the hot surface inside the guard and was connected to the automatic recorder *R* which was calibrated in millivolts. Provision was made for inserting glass plates in the holder *G* between the radiating surface and the pyrometer. The inverted position was used so that convection currents could

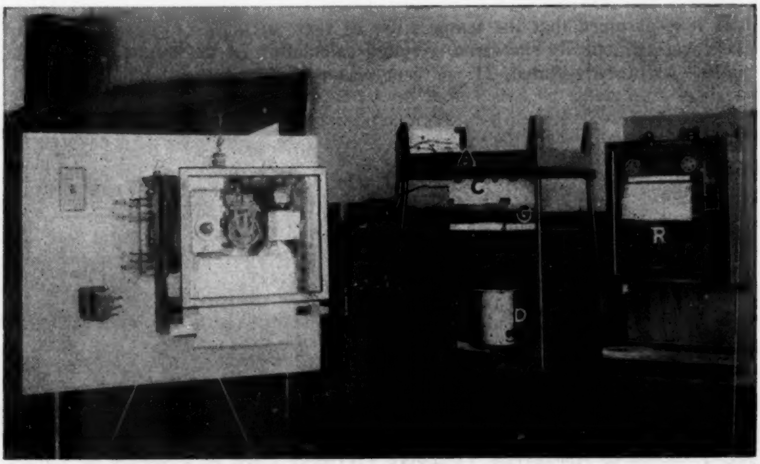


FIG. 6. VIEW OF TEST APPARATUS

not heat the glass and so that it would be affected only by the energy radiated from the hot surface.

After the radiating surface had reached a constant temperature (450 F), the recorder was allowed to run without any glass between the source and the pyrometer until a curve had been traced on the chart. This was of course a straight line as the temperature of the radiating surface was constant and indicated a constant deflection of 2.47 mv (millivolts). A single plate of glass ⅛ in. thick was now inserted between the pyrometer and the radiating surface (Fig. 7) and the deflection of the recorder at once fell to zero, showing that practically no radiation was passing through the glass. This seems to check the curves of Figs. 1 and 2 as derived from the Coblentz data. The deflection was actually less than 0.05 mv and was too small to be indicated by the recorder. As the radiant energy was absorbed by the glass, the top surface began to rise in temperature and this heat was of course conducted through the glass to the bottom surface and was re-radiated there as shown by a gradual rise in the curve traced by the recorder.

The pyrometer was now receiving the radiant energy from the glass instead of the hot surface. When the glass had reached an equilibrium in tempera-

ture, the recorder again traced a straight line. The constant deflection, however, was much less than before due to part of the original energy incident on the glass being reflected, part being radiated from the top surface and part of the energy of the hot glass being lost through conduction to the air touching the bottom and the top surface. The constant deflection of the pyrometer was 0.52 mv. Thus about 21 per cent of the original energy incident on the glass was transmitted or radiated from the bottom surface. This is shown on Curve V of Fig. 8.

INTERPRETATION OF RESULTS

If it is assumed that the temperature of both surfaces of the thin plate of glass were practically the same, a rough calculation of the total losses can be made. As already stated, 21 per cent was re-radiated from the bottom side,

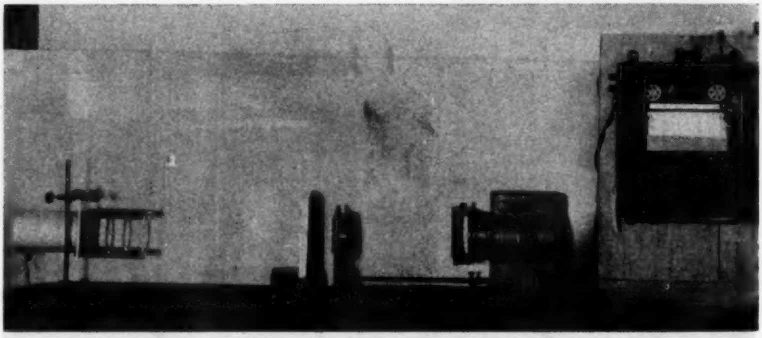


FIG. 7. VIEW OF APPARATUS, TUNGSTEN FILAMENT USED AS A SOURCE OF ENERGY

including the 1 per cent transmitted, and since a hot body radiates uniformly in all directions, 20 per cent would also be radiated back from the top surface. It will be assumed that 5 per cent was reflected back by the top surface and 1 per cent was transmitted as radiant energy by the glass. Of the total, 47 per cent has been accounted for and hence, 53 per cent was lost by conduction to the air from the bottom and top surfaces. If it is assumed that this conduction to the air was uniform on both isdes, the loss by conduction from the bottom surface amounted to 26.5 per cent which, added to the 21 per cent lost by radiation and transmission, amounts to a total loss of 47.5 per cent of the original energy from the bottom surface of the glass.

Curve VI of Fig. 8 shows the results of an exactly similar experiment, except that two plates of ⅛-in. glass were separated by a ⅜-in. air space. This experiment indicated that about 10 per cent of the energy incident on the upper glass surface reaches the pyrometer by radiation from the bottom surface of the lower plate together with that passing through by direct transmission. This is a greater loss than would be expected but it can be accounted for by noting that the bottom plate in this set-up is receiving all of its energy from radiation of the heated top plate as it can be assumed that the conduction through the air space is negligible. In the case of the single plate, the temperature of the glass

finally reached 220 F but with the two plates, the top plate reached a temperature of 275 F. This was because the heat loss by conduction from the bottom surface of the upper plate was stopped by the air space and also some of the energy radiated by the bottom surface was radiated and reflected back by the lower plate. Because of this increase in temperature, considerably more energy than the original 21 per cent was radiated from the bottom surface of the top plate. Part of this energy is reflected back by the lower plate as mentioned before and the remainder is absorbed by the lower plate, then part re-radiated back to the upper plate and the remainder lost from the bottom surface of the lower plate by radiation and conduction to the air. The lower plate reached a final temperature of 140 F.

As mentioned before, it was found that the amount of energy lost from the lower plate by radiation was 10 per cent of the total energy incident on the upper plate. Assuming that the amount of energy lost by conduction in each case was proportional to the absolute temperature difference between the air (80 F) and the glass, it was calculated that 11.5 per cent of the original energy was lost from the bottom surface of the lower plate by conduction to the air. Thus a total of 21.5 per cent of the total energy incident on the upper plate was lost from the lower plate to the outside. Since the total loss with a single plate was 47.5 per cent, the introduction of a second plate and an air space reduced this loss to less than half.

To show that this conservation was not due merely to the added thickness of glass, the two plates were replaced with a single plate of ½-in. plate glass. Although the thickness was twice as great as the combined thicknesses of the two plates, 16 per cent of the total incident energy was radiated from the bottom surface (Fig. 9) compared to 10 per cent in the case of two thin plates separated by an air space.

The foregoing calculations are not of course absolutely accurate but are intended to show only the approximate conditions. It should be borne in mind that these temperatures are considerably higher than those employed in heating human habitations, and therefore, that the proportion of transmissible wave lengths is considerably in excess of those encountered in household temperature conditions.

A significant feature of all of these curves is the fact that in every case, as soon as the glass was placed between the pyrometer and the source of energy, the deflection fell to zero. This clearly indicates that glass is opaque to radiations from bodies at low temperature and that radiant energy was again received by the pyrometer only after the glass itself was heated.

Due to the nature of the set-up, the only energy transfer between the hot surface and the glass plates was by direct radiation. If the set-up had been modified so that the energy could also have been transferred from the hot surface to the glass by convection currents, the losses through a single plate of glass would have been far greater.

Referring again to Fig. 8, curves IX and X show the actual results obtained when light at an approximate temperature of 3,000 C was used as the source of energy. The apparatus used is shown in Fig. 7. In this instance, it will be noted that the screening value of an increasing number of plates is more nearly proportional to the number of plates involved. There is very little of the

energy of visible light which is not transmissible by glass and in general, only that part is screened out which is reflected from the surfaces, since very little energy of these shorter wave lengths is absorbed by the glass.

The transfer of sensible heat through glass is, of course, subject to the same laws as in the case of any other material. It will depend upon conductances,

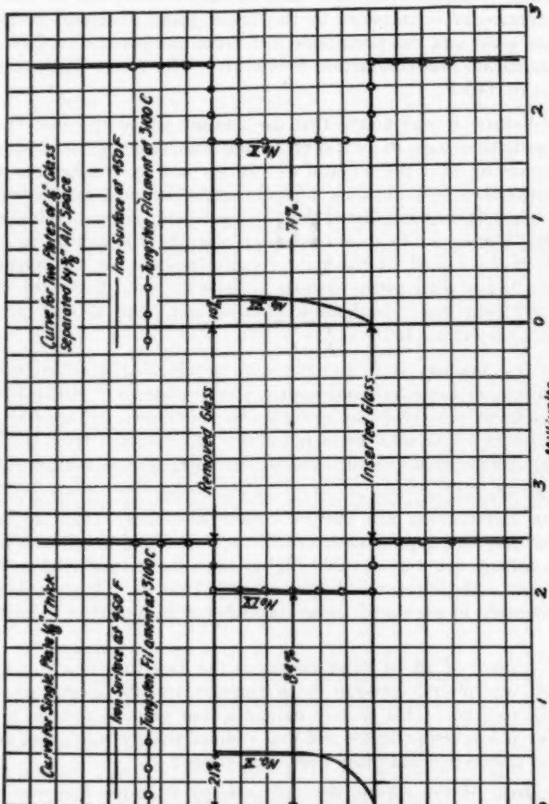


FIG. 8. CURVES DRAWN BY RECORDING PYROMETER PEN FOR SINGLE PLATE GLASS 1/8-IN. THICK AND TWO PLATES OF 1/8-IN. GLASS SEPARATED BY A 3/8-IN. AIR SPACE

film interchange as between the glass and the atmosphere, and vice versa, and the velocity of air flow over the glass surfaces. By calculation, based upon accepted determinations, it may be shown that three plates of 1/8-in. glass spaced approximately 1/2 in. apart are equivalent to a 12-in. brick wall[4] in

[4] See Chapter 3, A. S. H. V. E. GUIDE 1932. The coefficient of transmission of a 12-in. brick wall (no interior or exterior finish) is given as 0.295 Btu per hour per square foot per degree Fahrenheit difference in temperature whereas that for triple glass is given as 0.281.

insulating value. If this air space is reduced, the insulating value approaches that of an 8-in. brick wall.

The energy from the sun could easily enter through the glass to be absorbed and be retained within the room, but could not enter through a brick wall. That this heating by transmitted sun rays is considerable, is illustrated by a closed

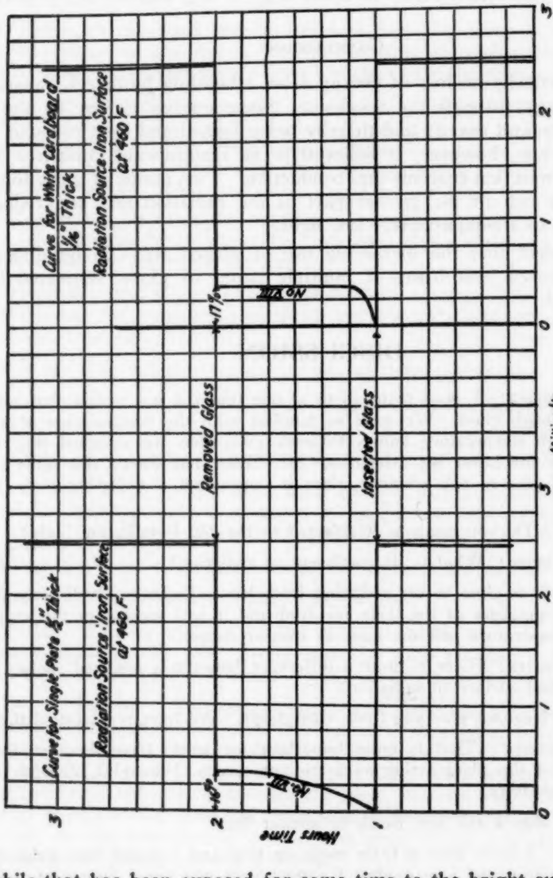


FIG. 9. CURVES DRAWN BY RECORDING PYROMETER PEN FOR SINGLE PLATE GLASS ½-IN. THICK AND WHITE CARDBOARD 1/16 IN. THICK

automobile that has been exposed for some time to the bright sunshine. Even when the outside temperature is zero, the inside of the car will be comfortably warm with no other means of heating the interior except by the transmission of the sun's rays through the glass.

Again in considering the transfer of sensible heat through glass, it is of course obvious that there is a maximum spacing between plates which will be

most effective. The maximum spacing for good results seems to be about ½ in. and the optimum spacing seems to be between ⅛ in. and ⅜ in. Preliminary investigations are being conducted along this line, and the insulating value of double glass with varying interglass spaces is being studied. It is interesting to note that these preliminary investigations indicate that there is very little difference between spacings of ⅜ in., ½ in. or ⅝ in., since the gradient shown under the same conditions seems to be 51 deg, 53 deg, and 50 deg respectively.

CONCLUSIONS

There is practically no loss of energy from a building by direct transmission of radiant energy through the windows. Some energy is lost by the glass absorbing the radiant energy and thereby being heated and then radiated to the outside. This loss, however, is believed to be insignificant compared to the losses due to convection currents and conduction. Two plates of glass separated by an air space cut off the greater part of the radiation loss and reduce the conduction loss to a comparatively low figure.

It seems evident that the increasing use of glazed areas in buildings need not mean increased heat losses if multiple panes of glass, separated by air spaces, are used.

DISCUSSION

PRESIDENT CARRIER: A good many of us at one time did not realize that radiation did not pass through glass. We were so familiar with the transmission of heat by sunlight and high temperature flames through glass that we assumed that it was proportionate for the lower wave lengths. Mr. Miller has shown this quite conclusively. I would like to ask whether glass is considered a radiating body in this paper?

R. A. MILLER: Do you mean as it referred to the 450 F radiating body?

PRESIDENT CARRIER: What is its coefficient of radiation?

MR. MILLER: It is close to the radiation from iron. Radiation is directly proportional to the temperature of the body involved and it will radiate in the same proportion to its temperature just the same as iron or copper.

PRESIDENT CARRIER: There is quite a difference between a polished surface and a rough or blackened surface in radiation.

MR. MILLER: I cannot give you those coefficients. We have not worked them out.

PRESIDENT CARRIER: That is quite important for us to have, because the conductivity effect of the glass acting as a radiant screen, I imagine, depends on the coefficient of emissivity.

MR. MILLER: May I ask Mr. Black to answer that?

L. V. BLACK: I have done a little work on that and I found that glass is very little lower than iron and iron oxide. I cannot give you exact figures.

F. C. HOUGHTEN: This paper gives accurately-measured quantitative data on a subject in which the Laboratory has been interested. The Laboratory has some qualitative data resulting from general observations in connection with other studies. In 1928 a paper was presented on heat radiation from the sun, and it was shown that double-strength window glass absorbed from 10 to 16 per cent of the radiant energy from the sun.

More recently, the Laboratory developed a pyrheliometer. If we use an incandescent tungsten light bulb as a source of radiation instead of the sun, a thin glass plate absorbs from 20 to 30 per cent of the radiation. If we use as our source of light a low-temperature electric heater, that is, a heater whose element is a cherry red, or, a temperature of 600 or 700 C, that same glass plate will absorb from 60 to 70 per cent of the total radiated energy. If we use as a source of radiation a black iron plate heated to steam temperature, the glass plate will cut out substantially all of the radiation.

These values are not quantitatively measured, but just observed in a qualitative way, but as near as we can observe them in that way, this rather thin clear plate of glass cuts out all the radiant energy from a surface at steam temperature.

MR. MILLER: I would like to point out one thing further, which is that in the comparison between a single plate of glass ½ in. thick and two plates of ⅛ in. glass, spaced ⅜ in. apart, the actual apparent transmission, or at least, the radiation from the colder surface of the ½-in. plate is materially higher than the radiation from the colder surface of the two plates placed with a space between.

I would point out also that the reflection value of the radiant energy from those surfaces plays an important part in the heating-up of the plate and the consequent re-radiation from the secondary surfaces.

L. A. HARDING: Mr. Miller, do you find the Stefan-Boltzman law applies in your case? What was the temperature to which this hot body was radiating? Did you use the temperature of the air?

MR. MILLER: The radiation pyrometer was at room temperature.

MR. HARDING: Did you find the Stefan-Boltzman radiation law holds true in your experiments?

MR. MILLER: Just exactly the same as the regular radiation curves.

MR. HARDING: There appears to be some question in regard to the Stefan-Boltzman radiation law for high temperature at the present time in high-temperature furnace work. That is the reason why I asked the question.

MR. MILLER: Those temperatures are in the upper ranges of high temperature.

MR. HARDING: Temperatures around 2,600 to 3,000 F.

MR. MILLER: That is way beyond the range at which we were working. The maximum temperature was 450. Temperatures of 2,600 to 3,000 F represent practically an incandescent body. There is a tremendous amount of radiation.

PROF. F. B. ROWLEY: This paper relates to a very essential problem in heat transmission. The question has often been raised as to whether heat is transmitted directly through glass by radiation without affecting the glass. It seems to me that data presented in this paper show that heat at short wave lengths and high temperatures is transmitted directly through the glass, but when these heat waves are absorbed by objects in the room and re-radiated, at low temperatures and long wave lengths, they do not pass directly through the glass. This is an important factor to be considered in summer cooling problems.

Mr. Harding brings up an interesting point about the Stefan-Boltzman law. It is possible that the law may be correct but the coefficients are different at higher mean temperatures. At the temperatures corresponding to short wave lengths, the coefficients which go with the law are reduced; in fact, at the very short wave lengths, they become practically zero. It may be the coefficients that have changed and not the law. The same condition has been found in heat transmission. We know that as we take higher mean temperatures within the range we work for transmission

through buildings, the coefficients change but the law remains correct. It seems to me that this point is worthy of further investigation.

PROF. A. P. KRATZ: I think one point that a great many engineers overlook is the fact that the Stefan-Boltzman law applies strictly only to a black body, or to a perfect radiator and to equilibrium conditions. It can be mathematically demonstrated only for a black body.

E. R. QUEER: In conjunction with the statements of Professors Kratz and Rowley, I would like to remark that it is the emissivity factor that modifies the Stefan-Boltzman law for radiations other than "black body." The emissivity is a variable factor depending upon the wave length at which the radiations are taking place. The Stefan-Boltzman law is considered quite accurate for all black body radiations. There has been some intimation that it may not hold at very low temperatures.

MR. HARDING: I think you are wrong in regard to the Stefan-Boltzman law. I believe it has been proved, for high temperature, that it apparently does not give correct results. Getting back to the question of concrete and the heat transmission tests in general, I am firmly of the conviction that we obtain two different results by the hot-box method of transmission and the use of the Nicholls heat meter. I believe we should run some parallel tests in the Laboratory and I am going to make this suggestion to the Director.

PRESIDENT CARRIER: I think that is a very good point, Mr. Harding. I believe that was touched on a little bit by Mr. Shodron. What are the temperature differences that are to be considered? With the Nicholls heat meter the differences between the surface of the body are largely considered while in the hot-box method, especially where there is relatively low air turbulence, there will be one temperature in the body of the air of the room, the surrounding walls, and another temperature as the surface is approached. This may vary the actual conductance over that calculated theoretically. I think that is a practical difference, where the difference you point out probably comes in.

MR. HARDING: The hot-box method of testing more nearly approaches the actual heating and ventilating installation, of course.

No. 917

IMPORTANCE OF RADIATION IN HEAT TRANSFER THROUGH AIR SPACES

By E. R. QUEER,[1] STATE COLLEGE, PA.

NON-MEMBER

HEAT is transmitted across confined air spaces by radiation, convection and conduction. It has been customary to regard each of these modes as an independent method but in this investigation it was found impractical to separate conduction and convection. These two modes may be conveniently designated by the term diffusion. Justification for this is based on the following conception of heat transfer in an air space: Part of the heat is transferred by radiation in accordance with the familiar Stefan-Boltzmann law; the remainder is transferred by conduction and convection which are inextricably combined. Heat leaves the hot surface by conduction through the surface film, then it is carried across the air space by a diffusion process which is a combination of conduction and convection to the receiving surface where it again passes through the surface film by pure conduction.

The factors affecting radiation in confined air spaces are the emissivity of the surfaces and their temperatures; those affecting diffusion are height and width of the spaces and the conductivity of the enclosed fluid. This paper summarizes the results of a comprehensive series of tests carried on at the Engineering Experiment Station of The Pennsylvania State College in which these factors were varied to determine their relative effect on heat transmission.

Although considerable work has been performed and reported upon heat transfer through air spaces, yet the importance of radiant heat transfer is still disregarded by many engineers. Additional data are herein presented to stress the importance of radiant heat transfer, and factors affecting conduction and convection in thin air spaces.

The electric hot plate method[2] was used to measure the thermal resistance

[1] Instructor in Engineering Research, The Pennsylvania State College, State College, Pa.

[2] See Bulletin No. 37, Heat Flow Meters and Thermal Conductivity Measurements, by F. G. Hechler and E. R. Queer (Engineering Experiment Station, The Pennsylvania State College).

Presented at the 38th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Cleveland, Ohio, January, 1932.

of the air spaces tested. The hot plate was placed between two similar test specimens and the heat transmitted was absorbed by cold water plates.

The effect of the size of air spaces was determined by using spaces varying in width from ⅓ in. to 1½ in. and in height from 4½ in. to 35 in. The effect of the nature of the surface on the radiation was determined by using polished

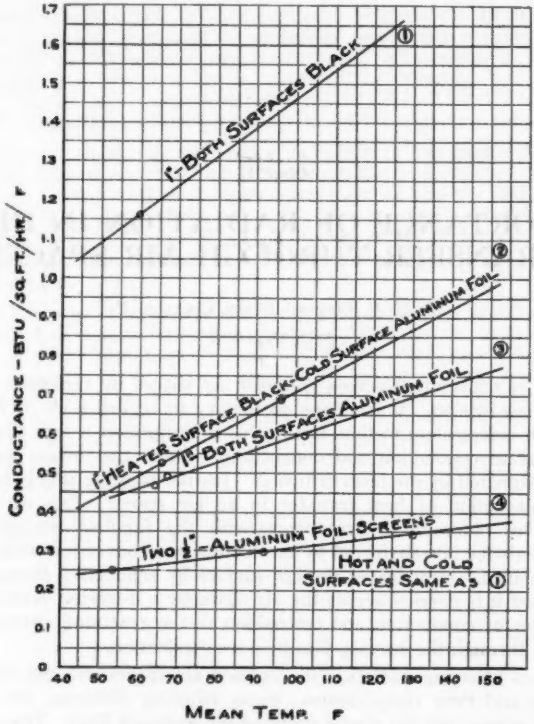


FIG. 1. EFFECT OF VARIOUS KINDS OF SURFACES ON HEAT TRANSMISSION THROUGH AIR SPACES

aluminum and aluminum covered with lamp black which has an emissivity about the same as ordinary building materials.

Both the hot plate and the cold plates were covered with aluminum foil (0.003 in. thick) cemented on with a thin film of raw rubber. The aluminum foil gave an excellent bright surface that did not change appreciably with time. Black radiating surfaces were obtained by painting the aluminum foil with lampblack mixed with alcohol. This coating could be rubbed off easily leaving the original polished aluminum surface unmarred.

Temperatures were measured with single thermocouples cemented to the surfaces with a waterproof cement. A comparison between this method of attach-

ing thermocouples and soldering them in a groove showed no appreciable difference in readings.

EMISSIVITY AND RADIATION

The effect of various kinds of surfaces on the heat transmission through air spaces is shown by the data in Table 1 which are plotted in Fig. 1. Curves 1, 2 and 3 furnish striking evidence of the efficacy of bright metallic surfaces in reducing heat transmission by reducing the radiation. Curve 4 shows the effect of placing one bright metal screen between two black surfaces. In this case there are two air spaces each one-half the width of the original space; this

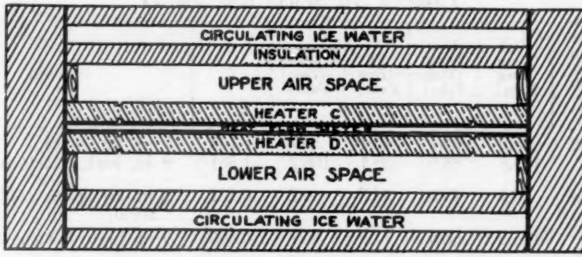


FIG. 2. APPARATUS USED TO DETERMINE E, SHOWING TWO HOT PLATES WITH HEAT FLOW METER BETWEEN

reduced somewhat the diffusional heat transfer by conduction and convection as well as the radiation.

To separate the total heat transfer into its component parts the Stefan-Boltzmann law was used to calculate the part transferred by radiation. The following equation gives the law in the form used:

$$H_r = \frac{0.172 \times \dfrac{e_1 e_2}{e_1 + e_2 - e_1 e_2}\left[\left(\dfrac{T_1}{100}\right)^4 - \left(\dfrac{T_2}{100}\right)^4\right]}{n+1} \qquad (1)$$

where

H_r = radiant heat transfer in Btu per hour per square foot
e_1 = emissivity factor for hot surface
e_2 = emissivity factor for cold surface
T_1 = absolute temperature of hot surface
T_2 = absolute temperature of cold surface
n = number of screens

$$E = \frac{e_1 e_2}{e_1 + e_2 - e_1 e_2} = \frac{e^2}{2e - e^2} \text{ (hot and cold surfaces alike)}$$

In the first series of experiments the combined emissivity factor E was determined. The bright aluminum surfaces of the heaters and cold plates were separated by one-inch air spaces as shown in Fig. 2. Two hot plates were placed one above the other in a horizontal position with a heat flow meter between

them. Heat was supplied to each air space by its respective heater. The heat input to the lower plate was held constant, while that to the upper plate was varied to keep the temperatures of the two heater surfaces facing each other equal. (This equality was indicated by a zero deflection of the galvanometer connected across the heat flow meter.) Hence all the heat of the lower heater test area was forced downward through the one-inch air space. By this arrangement convection was eliminated in this space, leaving only pure conduction and radiation. Deducting the heat transferred by conduction from the total heat input, the remainder was radiation. Substituting this heat transfer and the

TABLE 1. EFFECT OF VARIOUS KINDS OF SURFACES ON HEAT TRANSMISSION THROUGH AIR SPACES (See Fig. 1)

Curve No.	Heat Input (Btu per hr per sq ft)	High Temp. (Deg Fahr)	Low Temp. (Deg Fahr)	ΔT (Deg Fahr)	MT (Deg Fahr)	*C*	Remarks
1	17.59	67.2	52.1	15.1	59.7	1.163	1 in. air space,
1	55.5	123.7	88.6	35.1	106.2	1.581	9 in. high, black surfaces
2	17.64	83.1	49.3	33.8	66.2	0.522	1 in. air space, cold surface, alfoil,
2	39.7	125.5	68.0	57.5	96.7	0.690	9 in. high, hot surface black
3	17.59	86.1	50.4	35.7	68.3	0.492	1 in. air space,
3	39.50	135.7	69.1	66.6	102.4	0.594	9 in. high, hot and cold surface, alfoil
4	9.90	84.3	44.8	39.5	64.5	0.251	two ½-in. air spaces,
4	22.23	129.5	55.0	74.5	92.3	0.299	9 in. high, aluminum screen
4	40.25	189.6	72.1	117.5	130.8	0.342	hot and cold surface black

surface temperatures in Equation 1, E was calculated. A sample calculation follows:

Hot surface temperature = 155.4 F + 460 = 615.4 F, absolute
Cold surface temperature = 53.8 F + 460 = 513.8 F, absolute
Temperature difference ΔT = 101.6 F
Mean temperature (MT) = 104.6 F
Heat transferred by conduction through air = 0.172* × 101.6 = 17.49 Btu per hour per square foot
Total heat input = 26.15 Btu per hour per square foot
H_r = 26.15 — 17.49 = 8.66 Btu per hour per square foot
Hot and cold surfaces are alike, therefore $e_1 = e_2$

$$E = \frac{e^2}{2e - e^2}$$

$$= \frac{H_r}{0.172\left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right]}$$

* See Curve 1, Fig. 3. Conductivity of still air taken from the International Critical Tables.

$$= \frac{8.66}{0.172\left[\left(\frac{615.4}{100}\right)^4 - \left(\frac{513.8}{100}\right)^4\right]}$$

$= 0.0684$

$e = 0.128$ for these aluminum surfaces. These combined emissivity factors are tabulated in Table 2 and plotted as Curve 2, Fig. 3.

EFFECT OF SPACING

A second series of experiments was made to show how the conductance of vertical air spaces varied with width of the space. These results are plotted on Fig. 4. Two mean temperatures were chosen to indicate that the shape of the curve did not change over the range tested. It should also be noted that changing the height of the air space did not alter the shape.

TABLE 2. EMISSIVITY FACTORS

Heat Input (Btu per hr per sq ft)	High Temp. (Deg Fahr)	Low Temp. (Deg Fahr)	ΔT (Deg Fahr)	MT (Deg Fahr)	Heat Conduction (Btu per hr per sq ft)	H_r (Btu per hr per sq ft)	E	e*
16.65	116.0	46.8	69.2	81.4	11.57	5.08	0.0670	0.1256
26.15	155.4	53.8	101.6	104.6	17.49	8.66	0.0684	0.1283
44.53	220.7	67.5	153.2	144.1	27.90	16.63	0.0703	0.1310

* Aluminum used.

An examination of Curves 1 and 5 of Fig. 4 reveals an optimum single space conductance at ¾ in. spacing. This is in agreement with the results of other investigators. Curves 2 and 4 on this same figure show the effect of spacing on horizontal air spaces. These data represent an average conductance of equal spaces above and below the heater. In the upper space, convection is at a maximum whereas it is eliminated in the lower one. This is equivalent to considering the combined loss from the top and bottom of an insulated box, the insulation being made up of a series of bright surfaces. No optimum spacing for the single horizontal space was reached in the range tested.

EFFECT OF HEIGHT

Contrary to the general belief, it has been found that blocking thin air spaces decreases the insulating value of the space.[3] To show the magnitude of this effect a third series of experiments was made keeping the air space widths constant and varying the height from 4½ in. to 35 in. Tests at two mean temperatures were made to establish the position of a mean temperature-conductance curve. Fig. 5 shows a plot of some of these data for a one-inch space, and heights of 4½, 9, and 11⅝ in. The temperatures are also shown in this figure.

[3] See Testing Thermal Insulators, by H. C. Dickinson, M. S. Van Dusen (*A. S. R. E.*, Dec. 1915).

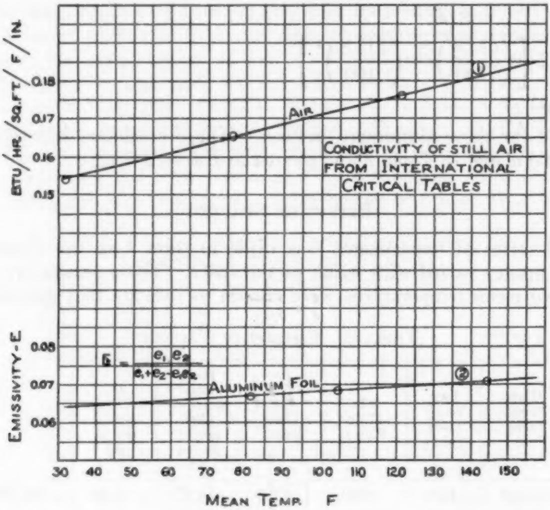


FIG. 3. CONDUCTIVITIES OF STILL AIR AND COMBINED EMISSIVITY FACTORS

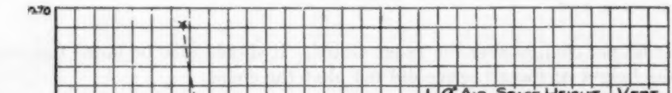


FIG. 4. VARIATION OF CONDUCTANCE OF VERTICAL AIR SPACES WITH WIDTH OF SPACES

If the temperature difference at any mean temperature is multiplied by the corresponding conductance for the same mean temperature the product is the total heat flow. The radiation may be calculated from the corresponding temperatures of the hot and cold surfaces and the combined emissivity factor. Subtracting this from the total heat input gives the diffusional transfer. Repeating this for each height and width of air space gave the values for the heat transferred by conduction and convection as plotted in Fig. 6. The following example illustrates the method of calculation:

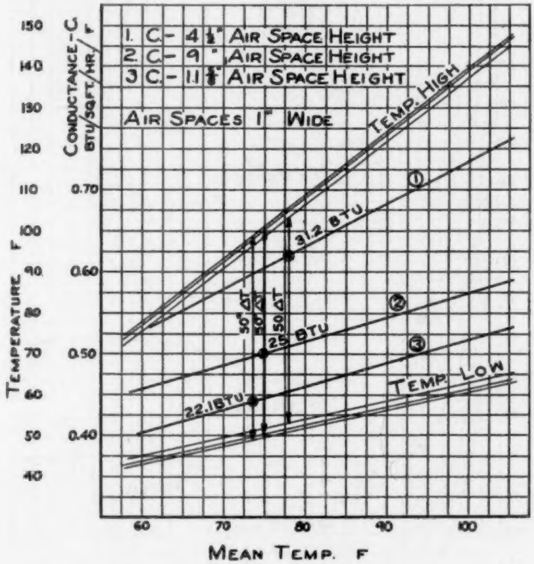


FIG. 5. MEAN TEMPERATURE—CONDUCTANCE CURVES

Example. A temperature drop between the hot and cold surfaces of 50 F was arbitrarily chosen. Height of air space, 9 in. Conductance (C) = 0.50 at a differential of 50 deg, MT = 75 F.

H (total) = 0.50 × 50 = 25 Btu per hour per square foot.

High side temperature = 100 F + 460 = 560 F, absolute.

Low side temperature = 50 F + 460 = 510 F, absolute.

E at 75 F (MT) = 0.0665 (From Fig. 3, Curve 2).

$$H_r = \frac{0.172 \times 0.0665 \left[\left(\frac{560}{100}\right)^4 - \left(\frac{510}{100}\right)^4\right]}{1}$$

= 3.54 Btu per hour per square foot.

H (diffusional) = 25 — 3.54 = 21.46 Btu per square foot per hour.

In like manner the other values of Fig. 6 were calculated. This figure is used merely to illustrate what will happen to the diffusional heat transfer at any temperature difference, while the height and width of spaces are changed.

This was a surprising phenomenon, since it has been the usual assumption that blocking thin air spaces decreased the conductance. The blocking was thought to decrease the fluid pressure differential between the hot and cold sur-

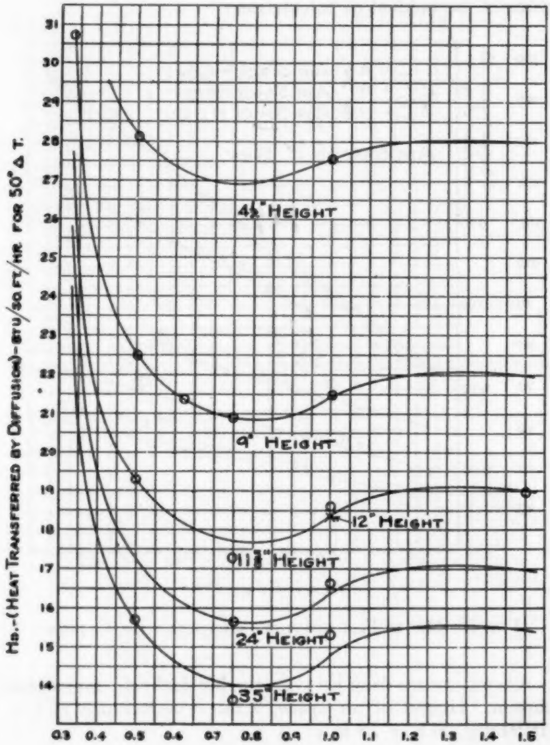


FIG. 6. CURVES SHOWING RELATION BETWEEN HEAT DIFFUSION AND AIR SPACE WIDTH

faces, consequently the diffusional heat transfer was decreased and the resistance to heat flow increased. Tests show that the reverse happened. Apparently what took place was that increasing the air space height increased the surface frictional resistance to air flow by an amount greater than the fluid pressure differential caused by differences in densities on the hot and cold surfaces. Hence, convection would be decreased and the resistance to heat flow increased.

The curve of Fig. 7 is a cross plot of Fig. 6 at one-inch spacing. The shape of this curve would be the same for any other spacing.

SUMMARY

It has been generally supposed that most of the heat lost through air spaces of building structures was by conduction and convection. However, this is not the case; most of the heat transferred between conventional building materials is by radiation.[4] Several examples are cited to show the magnitude of the radiant heat transfer between surfaces of the type mentioned.

Wall[5] *No.* 23

$\Delta T = 16.7$ F for air space between 2×4 studs

$T_2 = 49.8 \text{ F} + 460 = 509.8$ F, absolute

$T_4 = 33.1 \text{ F} + 460 = 493.1$ F, absolute

$$H_r = \frac{0.172 \dfrac{0.93^2}{0.93 + 0.93 - 0.93^2}\left[\left(\dfrac{509.8}{100}\right)^4 - \left(\dfrac{493.1}{100}\right)^4\right]}{1}$$

$= 12.65$ Btu per hour per square foot

Per cent of total heat transferred by radiation =

$$\frac{12.65 \times 100}{16.41} = 77 \text{ per cent}$$

If the 4-in. air space were divided into two 2-in. spaces by a thin bright metallic screen having an emissivity value 0.1, then

$$E = \frac{0.1 \times 0.93}{0.1 + 0.93 - (0.1 \times 0.93)} = 0.099$$

And if the same temperatures existed on the high and low surfaces

$$H_r = \frac{0.172 \times 0.099 \left[\left(\dfrac{509.8}{100}\right)^4 - \left(\dfrac{493.1}{100}\right)^4\right]}{2}$$

$= 0.72$ Btu per hour per square foot compared with a value of 12.65 Btu per hour per square foot for the original wall.

This same fact was borne out by the first series of experiments shown in Fig. 1, and accounts for most of the enormous reduction in the value of C between Curves 1 and 4.

Likewise for[5] Walls No. 25 and No. 30 respectively:

Per cent of total heat transferred by radiation =

$$\frac{11.32 \times 100}{15.10} = 75 \text{ per cent}$$

[4] See discussion by F. G. Hechler of R. H. Heilman's paper on Surface Heat Transmission. (*A. S. M. E.* Transactions—Fuels and Steam Power Division, Sept.-Dec. 1929. Vol. 51, No. 22, p. 299.)

[5] These data are taken from Heat Transmission Research, by F. B. Rowley, F. M. Morris and A. B. Algren (A. S. H. V. E. TRANSACTIONS, Vol. 34, 1928).

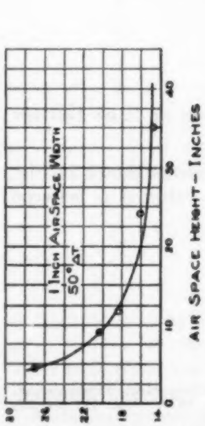


FIG. 7. CROSS PLOT OF FIG. 6 FOR ONE-INCH AIR SPACE

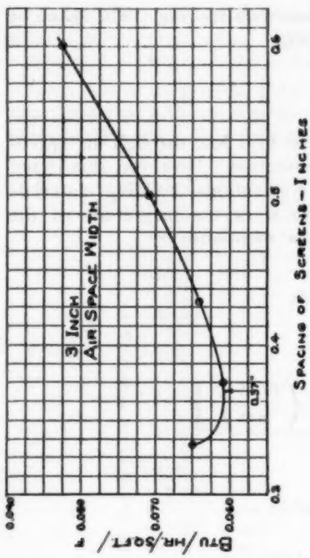


FIG. 8. OVERALL TRANSMISSION COEFFICIENTS FOR VARIOUS SPACINGS OF SCREENS

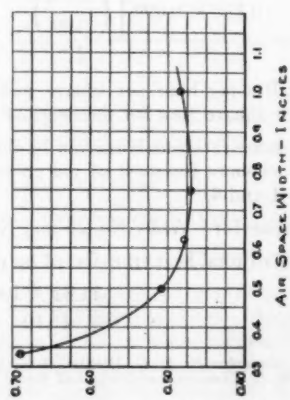


FIG. 9. CURVE SHOWING RELATION BETWEEN AIR SPACE AND WIDTH FACTOR

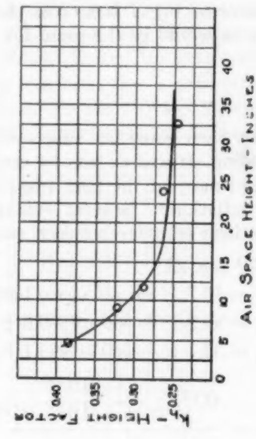


FIG. 10. CURVE SHOWING RELATION BETWEEN AIR SPACE HEIGHT AND HEIGHT FACTOR

Per cent of total heat transferred by radiation =

$$\frac{8.86 \times 100}{12.3} = 72 \text{ per cent}$$

This analysis indicates that approximately 75 per cent of the total heat transfer is by radiation. An optimum insulation spacing for multiple bright metallic surfaces that are very thin has been found to be 0.37 in. That is, if one has a given space to be insulated, the maximum insulating effect for a minimum number of bright metallic sheets is obtained by spacing sheets at 0.37 in. By choosing any space width and dividing it into spacings varying from about ⅓ in. to ¾ in., with bright sheets of aluminum one can calculate the approximate overall transmission factor for each spacing. Fig. 8 shows how this factor varies with spacing and indicates a minimum value at about 0.37 in. The following example will illustrate how a point on this curve was obtained:

Assume a 3-in. space to be insulated. A spacing of ⅓ in. would require 8 sheets of aluminum. Number of spaces $= 3 \times 3 = 9$. Height of space chosen, 11⅝ in. $C = 0.585$ Btu per hour per square foot per degree Fahrenheit (From Fig. 4). For practical purposes the conductance can be divided by the number of spaces, or

$$\frac{0.585}{9} = 0.065 \text{ Btu per hour per square foot per degree Fahrenheit.}$$

The curve of Fig. 8 was found to have the same shape for different heights and overall space widths. Some experimental data on multiple spaces are given in Table 3.

The effect of height of the air space on the heat transfer coefficient appears to become constant above 35 in. Spaces below 4½ in. have not yet been investigated.

Conclusions

1. Plane bright metallic surfaces form an excellent heat insulation by reducing radiant heat transfer.
2. Insulation made up of such surfaces is usually very low in density, and is not seriously affected by moisture.
3. The optimum insulation spacing for a single air space is about 0.75 in.
4. The optimum insulation spacing for multiple air spaces is 0.37 in.
5. Increasing the height of a thin air space up to 35 in. increases the resistance to heat flow.
6. With conventional building materials about 75 per cent of the total heat transmission across enclosed vertical spaces is by radiation.

Appendix

An attempt was made to calculate the heat flow through specimens tested from a knowledge of the emissivity factors, height, spacing and temperatures. This was found to be very difficult by dividing the heat flow into its three component parts, conduction, convection and radiation. However, a purely empirical equation was set up based upon diffusion and radiation, as follows:

$$H_T = \frac{k_w \times k_h \Delta T^{1.266}}{(n+1)^{1.1}} + \frac{0.172 \times \frac{e_1 e_2}{e_1 + e_2 - e_1 e_2}\left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right]}{(n+1)} \quad (2)$$

where

H_T = total heat transmission in Btu per hour per square foot
k_w = air space width factor (Fig. 9)
k_h = air space height factor (Fig. 10)
ΔT = temperature difference hot to cold surfaces, degrees Fahrenheit
Exponent 1.266 taken from Surface Heat Transmission, by R. H. Heilman
Exponent 1.1 was found to suit the experimental data the best
n = number of screens

TABLE 3. SOME EXPERIMENTAL DATA

Test No.	No. of Spaces	Space Width (In.)	Air Space Height (In.)	Heat Input (Btu per hr per sq ft)	High Temp. Side (Deg Fahr)	Low Temp. Side (Deg Fahr)	Temp. Diff. ΔT (Deg Fahr)	MT (Deg Fahr)	C (Btu per hr per sq ft per deg F)
1	1	½	9	17.58	83.0	47.2	35.8	65.1	0.491
2	1	½	9	39.55	130.2	60.9	69.3	95.5	0.571
3	1	½	11⅝	17.58	84.1	47.3	36.8	65.7	0.478
4	1	½	11⅝	39.55	134.7	62.3	72.4	98.5	0.546
5	2	½	9	17.58	116.8	49.0	67.8	82.9	0.259
6	2	½	9	39.55	195.7	63.8	131.9	129.8	0.300
7	2	½	11⅝	17.58	122.2	48.7	73.5	85.4	0.239
8	2	½	11⅝	39.55	214.0	64.3	149.7	139.6	0.264
9	1	⅓	9	17.58	75.9	47.9	28.0	61.9	0.627
10	1	⅓	9	39.55	117.9	61.9	56.0	89.9	0.706
11	3	⅓	9	9.9	88.1	44.2	43.9	66.2	0.225
12	3	⅓	9	22.25	145.0	55.0	90.0	100.0	0.247
13	3	⅓	11⅝	9.9	91.7	44.1	47.6	67.9	0.208
14	3	⅓	11⅝	13.49	107.9	46.0	61.9	76.9	0.218
15[a]	3	⅓	11⅝	17.58	97.0	47.0	50.0	72.0	0.351
16[a]	3	⅓	11⅝	39.55	156.9	57.5	99.4	107.2	0.398
17[b]	3	⅓	11⅝	17.58	132.4	49.0	83.4	90.7	0.211
18[b]	3	⅓	11⅝	39.55	226.0	63.4	162.6	144.7	0.243

[a] Tests No. 15 and 16 were made with 0.0003 in. crinkled aluminum foil.
[b] Tests No. 17 and 18 were made with 0.0003 in. plain aluminum foil.

TABLE 4. TESTS MADE TO CHECK EQUATION 2

No. of Spaces	Width of Space (In.)	Height of Air Space (In.)	Measured Heat Input (Btu per hr per sq ft)	High Temp. (Deg Fahr)	Low Temp. (Deg Fahr)	ΔT (Deg Fahr)	MT (Deg Fahr)	Calculated Heat Input (Btu per hr per sq ft)	Variation between Calculated and Measured Heat Input (%)
1	½	9	17.58	83.0	47.2	35.8	65.1	17.23	−2
1	½	9	39.55	130.2	60.9	69.3	95.5	39.62	+0.2
2	½	9	17.58	116.8	49.0	67.8	82.9	18.09	+2.7
2	½	9	39.55	195.7	63.8	131.9	129.7	41.6	+5.3
3	⅓	11	17.58	127.9	49.3	78.6	88.6	16.98	−3.4
3	⅓	11	22.25	145.1	51.3	93.8	98.2	22.25	−3.1

The second part of Equation 2 is the same as Equation 1.

The height and width factors were determined by taking the heat transferred by diffusion and temperature data for the one-inch space width from Fig. 7, and solving for k_h and k_w as a combined factor. By trial an arbitrary value for k_w of 0.48 was chosen and k_h was calculated for each height given on Fig. 7. Taking a height factor as a constant then k_w can be calculated for all widths. These two factors are plotted on Fig. 9 and Fig. 10.

This formula proved fairly satisfactory in calculating the total heat transfer in both single and multiple air spaces. An accuracy of 6 per cent was obtained by this method for the experimental range covered. Table 4 shows some tests made to check the formula.

DISCUSSION

W. J. KING (WRITTEN): This paper, which seems to be based upon some well conducted experimental work, should serve several useful purposes. Engineers in general seem to be inclined to underestimate the importance of radiation in low-temperature heat transmission, and the significance of this factor needs to be emphasized. There is also a tendency to depreciate the value of air spaces as thermal insulation, due to a lack of appreciation of the possibilities under the proper conditions.

Some authors have compared the conductance of several 1-in. air spaces, between surfaces of ordinary building materials, with that of cork or some other commercial insulant. Under these conditions the air spaces undoubtedly show up at a disadvantage. But if the air spaces are narrow, and are bounded by bright metallic surfaces, they may be more effective than any solid or porous insulating material on the market, as the present paper demonstrates.

The last column of Table 3 should be headed Btu per hour per square foot per degree Fahrenheit. Test No. 13 of this table gives the conductance of 3 spaces each ⅓-in. wide as 0.208. Since the total thickness is 1 in., this compares directly with the commercial conductivities of ordinary insulants, and it will be found that this figure is lower than the conductivity of any material listed in THE GUIDE.

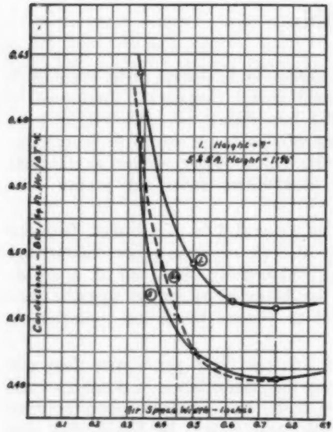
The most complete work on heat transmission through air spaces which the writer has seen is that of Mull and Reiher,[6] carried out at the Munich Technical High School in 1930. These authors found that the conductance of vertical air spaces decreased considerably at lower temperature differences. Their results indicate that the value of C in Test No. 13 might have been as low as about 0.18 if the temperature difference had been, say, 15 instead of 47.6 F. If the conductivity of the best commercial insulant available is assumed to be 0.23 Btu per hour per square foot per degree Fahrenheit per inch, air spaces under these conditions would be about 20 per cent better.

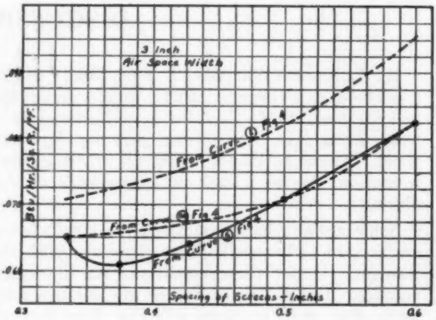
In Fig. 8, the author has indicated that the overall transmission of a 3-in. air space reaches a minimum when it is subdivided by screens at a spacing of 0.37 in., and increases as the spacing is diminished further. There seemed to be no reason for this, since the pure conduction is constant, for a given total thickness, and radiation and convection are continually suppressed as the spacing between screens is reduced. It was then observed that Fig. 8 was based on Curve 5 of Fig. 4, which might not have been drawn correctly. In the accompanying Fig. A, the broken

[6] *Der Warmeschutz von Luftschichten.* W. Mull and H. Reiher; Beihefte zum Gesundheits-Ingenieur, Reihe 1, Hft 28. Pub. by R. Oldenbourg, Munich and Berlin, 1930.

line, 5A, shows that another curve can be drawn through the 3 test points which determined this part of the original Curve 5. This new curve gives the result shown by the lower broken line of Fig. B, which indicates that there is no minimum point on the curve, and no optimum spacing for multiple spaces, as long as the screens are of negligible thickness. This is borne out by the similarity of the upper curve of Fig. B, which is based on the author's Curve 1, Fig. A. Conclusion 4 is therefore probably erroneous. There is no reason why the overall transmission of a fixed total thickness should not continually decrease as the number of interposed screens is increased, until the combined thickness of the screens themselves amounts to a considerable fraction of the heat path.

Fig. 6 shows the effect of height upon the heat diffusion across air spaces. In explaining this the author suggests that, "Apparently what took place was that increasing the air space height increased the surface frictional resistance to air flow





FIGS. A AND B

by an amount greater than the fluid pressure differential caused by differences in densities on the hot and cold surfaces." This inference is not compatible with observations of the convection process reported by several German and British investigators.[7] Careful measurements have shown that the air velocity over a vertical heated surface increases with the height. The heat transfer rate is less for tall surfaces because the air currents become rapidly warmed up as they rise along the hot surface, so that the effective temperature difference is reduced. At heights above 2 or 3 ft, depending upon the temperature, turbulence sets in, after which the coefficient is independent of the height.

A theoretical and experimental study of heat transfer in cylindrical gas spaces is reported by Wilhelm Beckmann, in the May, 1931 issue of *Forschung*. This includes

[7] Griffiths and Davis; *The Transmission of Heat by Radiation and Convection*. Special Report No. 9, Food Investigation Board, Dept. of Sci. and Ind. Research, London, 2nd edition, 1931.

E. Schmidt; *Experiments on Heat Transfer in Still Air*. Zeitschrift fur die Gesamte Kalte-Industrie, Nov. 1928, page 213.

Nusselt and Jurges; *The Temperature Field Over a Vertical Heated Plate*. Zeits. des V. d. I., May 5, 1928, p. 597.

Schmidt and Beckmann; *The Temperature and Velocity Fields Over a Vertical Plate Giving Off Heat by Natural Convection*. Technische Mechanik und Thermodynamik, Vol. 1, Nr. 11, p. 341 and Nr. 12, p. 391, 1930.

data on air, hydrogen, and carbon dioxide between concentric cylinders in the vertical and horizontal positions.

EDGAR C. RACK (WRITTEN): The author has contributed some interesting and instructive information on the subject of heat flow through air spaces. The data presented are worthy of careful consideration, particularly that dealing with the insertion of horizontal baffles in vertical air spaces and their effect on the resistance to the flow of heat across the air pockets.

Work was done some few years ago by the National Physical Laboratory of England which demonstrated that baffling air spaces will increase the heat loss across them. The results of these tests, however, were reported only in terms of arbitrary units and, therefore, have no particular value for practical application in calculations. The author is to be complimented for the determinations made and the data contained in this paper.

While it has been brought out that the effect of increasing the spacing of baffles will increase the resistance to heat flow, it is important to remember these tests were made on specimens prepared especially for laboratory use and were necessarily sealed so as effectively to prevent air leakage into and from the pockets. It is considered impracticable, if not impossible, with nearly all of the usual types of hollow wall construction to so erect them that they are effectively sealed against air leakage. Any small cracks or other openings in the wall will have a marked influence on convection losses.

I am a proponent of those types of wall construction which reduce air spaces to a negligible minimum and at the same time provide the desired insulating value. It is considered impracticable to apply bright metal foil barriers to break up the air spaces in building walls. As the other available types of material applicable for barriers add only a small percentage to the total resistance of the construction it would seem that the more logical course to pursue would be to fill the entire space with some material such as rock wool. In this way the influence of radiation across the air spaces will be effectively eliminated. At the same time the influence of convection will be reduced to a negligible minimum by obstructing the path of flow for convection currents to those cracks or other openings through the enclosing fabric where leakage occurs. When an efficient insulation fill is selected and properly placed in the air spaces, heat will be carried from the high to the low temperature surfaces largely by conduction only. The heat loss and leakage will, in such case, be a minimum.

With reference to data given in Table 3 it is of interest to note that Tests Nos. 15 and 16, made with crinkled foil, resulted in considerably higher values of conductance than when plain foil was used as barriers. In the case of crinkled foil, the heat flow is increased in proportion to the number and effectiveness of contacts between the individual sheets. Experience in our laboratory has demonstrated that the number and effectiveness of surface contacts between crinkled sheets is practically impossible of control within certain limits. We have had in continuous operation for a period of more than 6 months an apparatus for the determination of conductivity which is insulated with about 9 sq ft of crinkled foil. After constant conditions of operation are obtained, readings are taken and the temperature then changed to a higher or lower value by means of heater current adjustments. This cycle of raised and lowered temperatures has, to date, resulted in a conductance for 1 in. thickness of material ranging as high as 21 per cent above the minimum of 0.361 so far obtained at a mean temperature of 60 F. The indications are that the combined influence of weaving and vibration as well as expansion and contraction of the apparatus due to temperature changes is constantly varying the contact resistances between the crinkled sheets. The initial number of points of contact as well as the contact resistances may be considered as variables which will be dependent largely upon the personal factor of the individual applying crinkled sheets.

It is a matter of interest to note that the author has not expressed the results of his tests in terms of thermal conductivity. The writer feels that the conductance factor as used by the author is the proper term in which to express the thermal properties of assembly types such as are considered here. We do see many references to the specific quality of thermal conductivity of foil-air assemblies, and while the term "apparent conductivity" might properly apply for the foil-air types, there is no reason why homogeneous and assembly types of various material cannot be compared on the basis of their conductance values if all are expressed in terms of even thickness.

In closing it is suggested that the author include in this report, as a matter of record, the ratio of the guard ring width to the length of one side of the main heater as it applies for the hot plate apparatus used in his various tests. This report should also include the ratio of guard ring width to the maximum thickness of the construction types tested. Experience in our laboratory has denoted that this last ratio should be not less than 1.0. In testing specimens of relatively thin section and small area, our practice is to increase this ratio. Our tests on assemblies made up of combinations using relatively large volumes of air further indicate that a higher ratio than 1.0 will probably give more dependable results. The data supplied with that reported, will be of service to other investigators in their future reference to this contribution by Mr. Queer.

E. B. SVENSON (WRITTEN): The data presented are a definite contribution to our knowledge of heat transmission, particularly with respect to the performance of bright metallic surfaces in the field of heat insulation.

Although the effectiveness of polished metal screens in preventing heat radiation has been known for a long time, it has been of very little advantage in the insulating field until recently. Even yet few of us realize the importance of radiation in the transfer of heat across air spaces. Fig. 1 of Prof. Queer's paper illustrates this quite effectively as well as his analyses of several typical walls.

Another pill which is hard to take is that there is no advantage, from the insulating standpoint, in restricting the height by means of partitions of thin vertical sections of air bounded by metallic surfaces. This was well illustrated by an experiment made at the Aluminum Research Laboratories where panels made up of foil spaced by corrugated paper transmitted more heat when the corrugations were horizontal than when they were vertical.

One item of the report to which attention should be given in view of the general interest in aluminum foil insulation at the present time is the conductance across a 1-in. air space divided by two crinkled foil screens, as given in Tests Nos. 15 and 16, Table 3. There is no question as to the accuracy of the author's results for the particular condition, but it is possible to secure much better results by the use of 3 screens of properly crinkled foil in a 1-in. air space. The Aluminum Research Laboratories obtained conductances of 0.29 to 0.305 Btu at 72 F mean temperature and 0.31 to 0.335 Btu at 107 F mean temperature under similar test conditions across 1-in. air spaces divided by 3 layers of foil crumpled according to recommended procedures.

We make this comment to prevent any misconception of the excellent insulating value of crumpled aluminum foil and hope it does not detract from the heartiness of our endorsement of Prof. Queer's excellent work.

P. NICHOLLS [8, 9] (WRITTEN): The author is to be commended on the work he presents, and I believe that, for the imposed conditions, the numerical values he gives are nearer the truth than those of previous investigators. The paper, however, has the same weakness as have the majority of such reports of investigators on heat

[8] Published by permission of the Director, U. S. Bureau of Mines. (Not subject to copyright.)
[9] Supervising Fuel Engineer, U. S. Bureau of Mines, Pittsburgh, Pa.

transfer: namely, that it does not include the checks to confirm the order of accuracy of instruments, apparatus, methods, and results. The extending of investigations to obtain such checks and the including of them in the report is of the greatest importance; otherwise, as far as the values reported differ from those of other investigators, there is a lack of certainty as to which are the more reliable. It would be a pity if the author should fail to extend his work, as he has the opportunity to obtain data which are close to fundamentals.

The main omission is that the paper neglects the fact that if the area of the air space—that is, its length and height—be finite, then the heat transfer by the bounding edges can not be neglected, and it is highly important at least to include full data on their construction. Not only do they carry heat by conduction but, for wider air spaces, they will affect the radiation; an exact analysis of this action would be quite complex. The objective should be to have the data in such a form that when computing the transfer one could allow for the spacers used in the construction being considered.

The spacers around the edges of the plate as indicated by Fig. 2 were ½-in. wood, and from the dimensions given for the subdivided heights it would look as if the other spacers were also ½ in. However, in the absence of information of what they were and how the author took care of the area they occupied, and the heat transfer through them, it is not possible to comment on the large change in heat transfer with height of space.

Several deductions in the paper depend on the accuracy of the value determined for the emissivity, and it would be worth while to obtain values for both the bright and the black aluminum surfaces and for their combinations by varying the conditions. One such variation might be using other widths than 1 in. for the air space. Presumedly the values of Table 2 were obtained by using the 35 × 35-in. area. Were the surfaces of the bounding wood spacers covered with foil? Since the heaters used were presumedly of the guarded type, there will be some question on the correct value to take for heat input and assigning it to the area under consideration. It would be well to repeat the work using the 24-in. guarded area and cork spacers faced with aluminum foil. A cork spacer ¼ in. wide should be substantial enough and, if desired, it might be possible to confirm the estimated value for the heat transfer through the spacers by using more than one width. The conductivity of the material used for the spacers should, of course, be determined.

The order of accuracy is dependent on those of the rate of input and of the temperature measurements. In this method of employing a heat flow meter to obtain unidirectional flow as originally developed by myself for the calibration of heat flow meters,[10] I made the balancing meter of ½-in. cork to obtain greater sensitivity. I also divided the couple circuit into sections to measure the equality of temperature over halves or quarters of the area. If there is only one system of couples, all one knows is that there is an average balance, but subdividing the couples usually shows that there is flow between heater plates in one direction over part of the area and in the other direction over another part. This unbalance can be much reduced by adjusting the pressure between the heaters.

These questions on accuracy should not be interpreted as necessary to applications of air space transmission in building construction, in which precise knowledge of conditions is usually lacking and in which air leakage or infiltration is often probable. They suggest rather the need of accurate knowledge on fundamentals, although the same order of accuracy would be applicable to built-up box insulation; again, however, since such insulation will usually be for refrigerating apparatus, the question of condensation as affecting heat transfer can not be overlooked.

The statement is not correct that the usual assumption has been that reducing the

[10] A. S. H. V. E. TRANSACTIONS, Vol. 30, 1924, p. 65.

height of the air space decreases the conductance. Dickensen and Van Dusen [11] showed a large increase for an 8-in., as compared with a 24-in. high air space. Griffith and Davis [12] show no change for wide spaces by halving the height. Kreuger and Eriksson,[13] experimenting on double glass windows with ⅝-in. space, show an increase in overall transmission when spacers are used, but obtain the same increase when the spacers are placed vertically. No one has separated the spacer transmission from the total. Acceptance of the diffusion values shown by the author in Fig. 6 should therefore be withheld.

R. E. BACKSTROM (WRITTEN): Everyone who has read Prof. Queer's paper must have found much interesting and some rather startling information. Referring first to that portion of the report dealing with the width of air spaces, it is interesting to note that a width of approximately ¾ in. was found to be the optimum dimension. As further evidence of this contention, it is only necessary to refer to other investigations along this line. In 1926, Professor Rowley presented before this Society a paper entitled, Some Results of Heat Transmission Research, in which it was pointed out that in increasing from ⅛ in. to 2½ in. the space between two panes of window glass, "the best condition was reached at about ¾ in. spacing." Professor Rowley employed the hot-box and the section tested was 36 in. square, whereas Prof. Queer used the hot-plate and the areas tested were 9 in. and 11⅝ in. square. The fact that the results from both investigations were in such close agreement in spite of the different methods of testing and the different height of test sections would seem to substantiate Prof. Queer's statement that changing the height of the air space did not alter the shape of the curve, from which we can conclude further that so far as total heat transfer is concerned, ¾ in. air space is apparently the optimum width regardless of height.

If, however, we consider separately the various modes of heat transfer, we find that the height of an air space has a marked effect (but not the effect we would normally expect, as the author has discovered) on that portion transmitted by convection. This is one of the most interesting features of the report, and no one can deny that the results are contrary to general opinion. It seems paradoxical that an air space 35 in. high should offer more resistance to the convection mode of heat travel than one 4 in. high, yet the author has produced some convincing evidence to show that this is the case.

The curve in Fig. 7, showing the amount of heat transferred by conduction and convection across a 1 in. air space of varying height, suggests that little, if any, further decrease will result from increasing the height beyond 40 in. This conclusion apparently holds true for widths of 1½ and 2 in., the limit of these tests. But what happens in spaces in excess of 2 in.? At some width, will not the fluid friction between air particles, in the increased volume, be less than the friction between air and surface of material, and result, therefore, in convection in the central section of the air column? In this case, height obviously would affect the amount of convection. It would seem that further tests would be warranted to establish the relation between width and height, particularly with respect to convection.

The author uses the term "diffusion" in speaking of the heat transferred across an air space by conduction and convection. Mr. Houghten and his associates at the Research Laboratory use the term "diffusivity" in their paper on, Heat Transmission as Influenced by Heat Capacity and Solar Radiation to describe a certain property of a homogeneous material which varies with conductivity, specific heat, and density. It would seem that much needless confusion will result from the use of these similar terms. Furthermore, I do not see the need for the term "diffusion" which

[11] Journal American Society Refrigerating Engineers, Vol. 3, Sept. 1916.
[12] Department of Scientific and Industrial Research, Food Investigation Board, Special Report No. 9.
[13] Royal Academy of Scientific Industrial Research, Sweden, Report No. 7.

Prof. Queer has adopted, but if it is felt desirable to designate the combined effect of conduction and convection by a single word, I would suggest that the author and the Society give consideration to some other term that will not be mistaken for "diffusivity" which has a definite physical and mathematical significance.

With the bounding surfaces at or below room temperature, it is surprising to note the large proportion of the total heat transmittance which results from direct radiation in walls containing large air spaces.

President Carrier: I think this is a most interesting paper, pointing out the importance of radiation. Did you ever think why heat was obtained from an attic space to a lower floor, as an example, in summer when cooling the room below? You can figure the conductivity of the air in that space and the heat transmission by that method as negligible. There is no convection because the hotter surface is above rather than below. The heat that is transferred is by radiation and it is about half as much if the direction of heat flow is reversed, and it is a considerable amount that cannot be eliminated except by methods outlined by Prof. Queer, by radiation shield or equivalent insulation.

Prof. F. B. Rowley: In considering an over-all heat transmission coefficient, we must take into account surface conductance, conductivity of the materials involved, and air space conductance, each of which is affected by several factors. Mistakes have been made in the past by not making a thorough analysis of the problem. As the last speaker stated, much of the insulation in the past has not been designed. In the future, I believe the results of present research work will be profitably applied to insulation at all temperature ranges.

The results given in the paper agree in the main with those which we have obtained at Minnesota. We have not found, however, as much of a variation due to the height of the air space as Prof. Queer has found.

Mr. King: In a good many cases a great deal of expense and time can be saved by studying the literature of what has been done already in this field.

For example, there is a great deal of activity in Germany at present. There are 4 or 5 German and English papers on the subject and I might say they all bear out in general the results of the author.

E. R. Queer: In answer to the comment regarding the minimum multiple spacing of sheets of foil as being at 0.37 in.; this was modified in the presentation of the paper. The experimental values did not check the theoretically determined value. However, we did find a minimum point in our experimental work at 0.28 in., for the foil we used. The following conductances per inch were found at a mean temperature of 110 F:

Multiple spacing, inches	*Conductance per inch*
0.25	0.24
0.28	0.23
0.50	0.24

It is of utmost importance in any research project to carefully examine the literature. As pointed out in the paper much of these data have been available in some form or other. Seventeen years ago the Bureau of Standards showed that increasing the height of thin air spaces increased their resistance to heat flow. Other data date back 50 years, but regardless of these facts, engineers continue to disregard the importance of radiant heat transfer at low temperatures.

A good point brought out in one discussion is that filling the air space eliminates radiation. This is true; however, there is no fill material of which I have any

knowledge that will produce conductances per inch as low as that of bright metal foil under certain conditions of temperature.

In all the tests made the guard ring width to specimen thickness ratio was greater than one.

Mr. Nicholls calls attention to the fact that calibration of instruments and apparatus are not included in the paper. Should all this material be included; it would have made quite a voluminous paper. However, from time to time the calibration of all instruments and apparatus is checked in a standard instrument laboratory.

Radiations to and from spacers and cardboard strips were minimized by covering them with aluminum foil and the effect upon the results was negligible. When it was necessary to divide the areas to change the heights of the air spaces, 3/64-in. cardboard strips were fitted into the grooves surrounding the test area on the guarded heater. These were replaced by 1/16-in. cork strips and no appreciable variation in results was observed.

It should be remembered that spaces thicker than 1 in. were used only once. Most of the data were taken on spaces of 1 in. and less.

Aluminum foil was used on account of the permanence of the surface during the test. Any other bright metallic foil would have served our purpose just as well providing the emissivity remained constant during the tests.

No. 918

HEAT EMISSION FROM IRON AND COPPER PIPE

By F. C. Houghten [1] (*MEMBER*) and Carl Gutberlet [2] (*NON-MEMBER*)
Pittsburgh, Pa.

MANY studies have been directed towards the determination of heat emission from bare and covered standard iron and steel pipe carrying steam and water at various temperatures. This paper presents data on the heat emission from bare copper pipe under service conditions, resulting from a study made at the Research Laboratory of the American Society of Heating and Ventilating Engineers, in co-operation with the *Associated Copper Tubing Manufacturers.* A few data are included giving heat emission for galvanized, brass and aluminum pipe, the effect of painting the pipe on the rate of heat emission, and the reduction in heat emission from standard iron pipe resulting from the application of commercial insulation.

Since heat emission from iron pipe has been the subject of much research and analysis, and since the present study on copper tubing was of necessity brief in comparison, the study of heat emission from copper tubing was made comparative with heat emission from iron.

Test Set-Up and Procedure

All tests were made in the psychrometric chambers of the Laboratory in the Pittsburgh Experiment Station of the United States Bureau of Mines where surrounding atmospheric conditions could be accurately controlled. This room is located in the interior of the building with well insulated walls.

The inside wall surfaces in view of the test pipe were therefore at the same temperature as the surrounding air.

Fig. 1 is a photograph of the set-up used for studying horizontal pipe. Figs. 2 and 3 are drawings of the arrangement of the apparatus for studying vertical and horizontal pipe respectively. Steam, at a constant controlled pressure, held well within the limits of 0.1 in. of water column, entered the steam separator,

---

[1] Director, Research Laboratory, American Society of Heating and Ventilating Engineers.
[2] Research Assistant, Research Laboratory, American Society of Heating and Ventilating Engineers.

Presented at the 38th Annual Meeting of the American Society of Heating and Ventilating Engineers, Cleveland, Ohio, January, 1932, by F. C. Houghten.

A, at *B*, and passed through the electric superheater, *C*, to the two test pipes, *D* and *D'*.

The condensate was drained from the test pipe at *E* and *E'* through the water sealed traps, *F* and *F'*, and was collected in graduated cylinders, *G* and *G'*. The tops of the graduate cylinders were covered so as to lessen possible re-

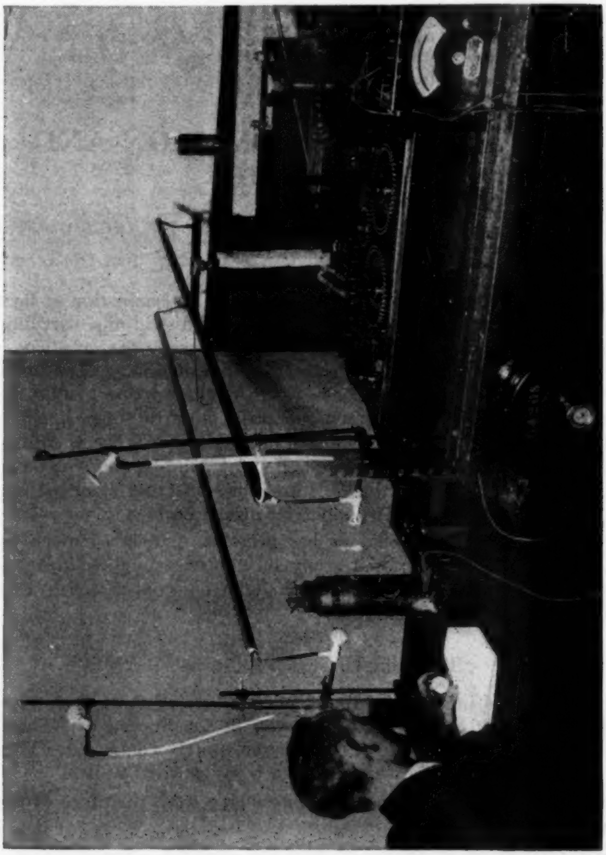


FIG. 1. LABORATORY SET-UP FOR DETERMINING HEAT EMISSION FROM HORIZONTAL PIPE

evaporation from the receiver. A small thread of steam was allowed to escape at *H* and *H'* throughout the test in order to keep the system free from air.

Before starting a test the entire system was purged of air, and heated for a sufficient length of time to insure equilibrium conditions. The valves, *I* and *I'*, were adjusted to give equal steam pressure in both pipes. The electric current through the superheater was regulated to maintain approximately 0.1 deg of

superheat in the entering steam as indicated by thermocouples located at *J* and *K* and *K′*.

Throughout the test, temperatures were observed at points *K, L, M, N,* and *O*. Thermocouples *K* and *O* were located in ⅛-in. O.D. copper tubes, with the junctions in the steam inlet to, and the drain from, the pipe. The outside surface temperatures of the test pipes were observed at *L, M,* and *N* by thermocouples whose junctions were soldered into holes drilled part way through the walls of the test pipes, care being taken to allow the solder to cover as little of the pipe surface as possible. All thermocouples were of No. 40 B & S gage wire.

During the test, the water level in the water-sealed trap was maintained at a level 0.5-in. below the top of the glass section *P* of the trap to reduce the steam heated surface other than the pipe itself to a minimum.

The volume and temperature of the water in the receivers were observed and plotted every ten minutes, and all tests were continued until two consecutive periods of one hour showed the same condensation, or until the curve resulting from the 10-min observations gave a continuous straight line. The volumetric determination of condensation was checked by weighing at the beginning and end of each test.

The ends of the pipe were closed by discs soldered in place. The steam supply drain and air vent connections were of small copper tubing soldered in place. Samples of pipe 9 ft in length were studied. In the vertical position, the pipes reached from 8 in. above the floor to 6 in. below the ceiling, and were separated from each other by 30 in., and from the nearest wall by 30 in. The pipes studied in the horizontal position were 30 in. apart, pitched 2 in. in the 9-ft length, and were supported by fine wires. In the horizontal position, they were located 24 in. or more from any wall, and 48 in. from the floor. The necessary instruments, other equipment and the observer were located so as to offer no interference to normal convection currents.

DATA AND RESULTS

All tests were made with steam temperatures ranging from 212.3 F to 213.5 F. The entering steam temperature measured at *D* and *D′*, Figs. 2 and 3, was always from 0.3 to 0.5 deg higher than the temperature observed at the drains *O* and *O′*. Since the entering steam contained about 0.1 deg superheat, the steam temperature as measured at the steam supply and the drain ends of the pipe varied by less than 0.4 deg. No doubt some of the observed difference was due to heat conduction through the water sealed trap from the lower end of the pipe. The steam temperature as taken from steam tables [8] for the observed steam pressure and barometric pressure was always about 0.3 deg higher than the observed entering steam temperature. The entering steam temperature less the degree of superheat was accepted as the steam temperature for heat loss computations.

The three thermocouples attached to the outside surface of the pipe invariably showed a temperature from 0.3 deg to 0.5 deg lower than the accepted steam temperature. A part of this difference is necessarily due to the temperature

[8] Steam Tables and Mollier Diagram, by Joseph H. Keenan.

drop through the pipe wall. Because of the conduction of heat away from the surface through the fine thermocouple wires, the true temperature drop through the wall is less than this observed difference. For the purpose of calculating

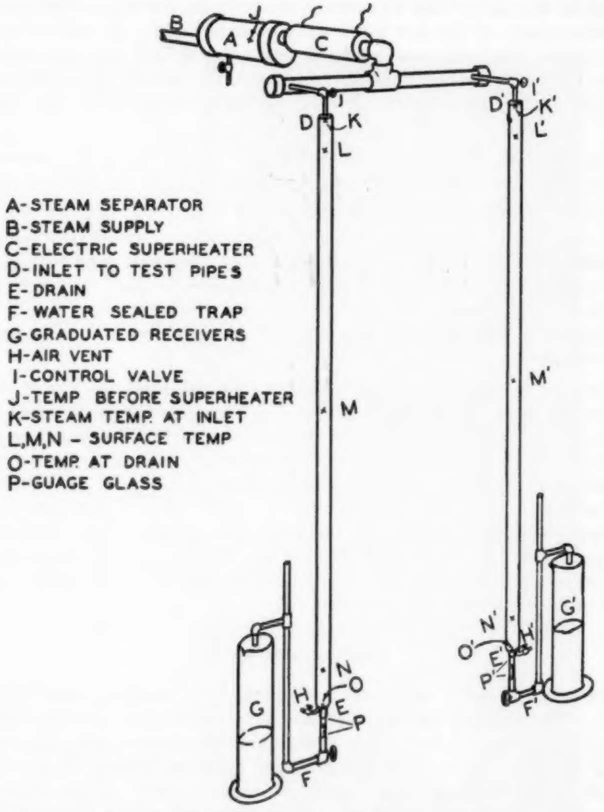


FIG. 2. TEST SET-UP FOR VERTICAL PIPE

heat loss coefficients, the pipe surface temperature was assumed to be the same as that of the steam.

The following two coefficients were calculated for the bare pipe:

1. The thermal transmittance coefficient (U) which is the heat emitted in Btu per hour per square foot of surface per degree temperature difference between steam and air. Since the steam temperature was practically that of the pipe surface, this coefficient is also the film transmittance coefficient for the

outside surface of the pipe. The thermal transmittance coefficient (U) is given by the formula:

$$U = \frac{r\,W}{A\,(t_s - t_a)} \tag{1}$$

2. The heat emission from the pipe (H) in Btu per hour per foot length of pipe per degree temperature difference between the steam and air.

$$H = U \times \text{(square feet of pipe surface per linear foot of pipe.)} \tag{2}$$

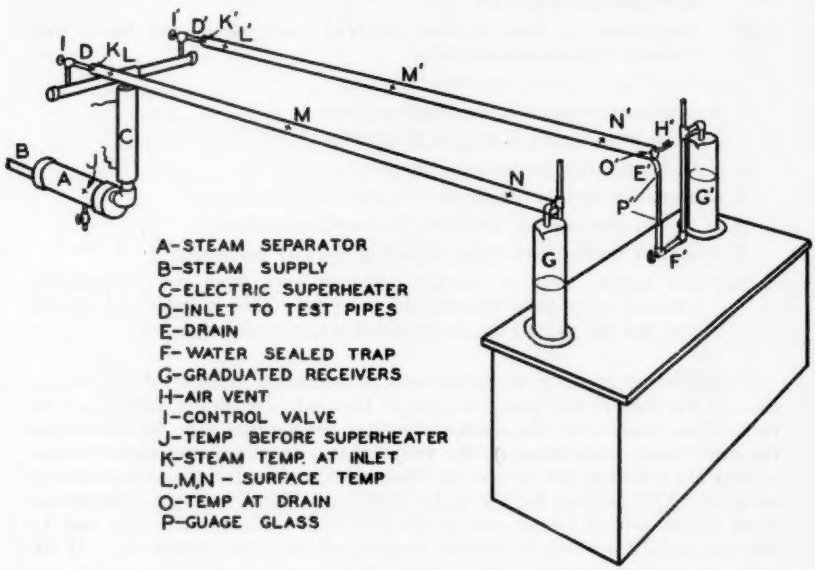


FIG. 3. TEST SET-UP FOR HORIZONTAL PIPE

The following two coefficients were calculated for the insulated pipe:

1. The film transmittance coefficient (h_I) which is the heat emitted in Btu per hour per square foot of insulation surface per degree temperature difference between the surface of the insulation and the air. It is given by the formula:

$$h_I = \frac{r\,W - U \times \text{(auxiliary surface area)}}{A_I\,(t_I - t_a)} \tag{3}$$

2. The heat emission through the insulation (H_I), in Btu per hour per foot length of insulated pipe per degree temperature difference between the steam and air.

$$H_I = \frac{r\,W - U \times \text{(auxiliary surface area)}}{A_I\,(t_s - t_a)} \times \text{(Square feet insulation surface per foot length of pipe)} \tag{4}$$

where

U = transmittance coefficient steam to air expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit temperature difference steam to air.

H = Btu emitted per hour per foot length of bare pipe per degree Fahrenheit temperature difference steam to air.

h_1 = film transmittance coefficient for the insulation surface in Btu per hour per square foot of insulation surface per degree temperature difference insulation surface to air.

H_1 = Btu emitted per hour per foot length of insulated pipe per degree temperature difference steam to air.

r = latent heat of steam at temperature t_s.

W = weight of condensate in pounds per hour.

t_s = steam temperature in degrees Fahrenheit.

t_a = air temperature in degrees Fahrenheit.

t_1 = insulation surface temperature in degrees Fahrenheit.

A = area of pipe surface, including the auxiliary surface.

A_1 = area of insulation surface, including ends of insulation.

Auxiliary surface area = surface emitting heat from the condensation weighed other than the sides of the pipe. This includes ends of the pipe and the part of the water sealed trap containing steam.

A small error exists in the coefficients as calculated, because of the surface area of the ends of the pipe, the part of the drain containing steam, and the ends of the insulation. The auxiliary surface for the bare pipe did not include the steam supply connection for the reason that it contained superheated steam; neither did it include the air vent as steam condensed in this connection was not weighed. This surface for the ¾-in., 1-in., and 1¼-in. bare pipe, constituted from 1.5 per cent to 1.8 per cent of the total bare surface of the pipe, and 1.9 per cent and 2.2 per cent of the 4-in. copper and iron pipe respectively. If this auxiliary surface emitted as much heat per unit area as the sides of the pipe there would be no error. It is probable that this surface emitted less heat per unit area than the test pipe, and therefore, the error due to the extra surface was considerably less than the percentage this surface is to the surface of the entire pipe. The ends of the insulation constituted 1.2 per cent to 1.3 per cent of the side surface area. Since the heat loss per unit area of the ends is probably less than that of the sides, the resulting error in the coefficient should be considerably less than 1 per cent.

Table 1 gives the heat emission in Btu per hour per square foot of pipe surface per degree temperature difference between steam and air, and the Btu per hour per foot length of pipe per degree temperature difference between steam and air, as determined by the A. S. H. V. E. Laboratory for the bare iron pipe. The table also gives the heat emission for bare-iron pipe as reported by Heilman.[4] The last two columns of the table give the surface transfer coefficients for the insulation surface in Btu per hour per square foot of insula-

[4] Heat Transmission from Bare and Insulated Pipes, by R. H. Heilman (*Industrial and Engineering Chemistry*, May, 1924).

Table 1. Heat Emission from Bare and Covered Iron Pipe

Position	Size (Inches)	Bare			Covered	
		A. S. H. V. E. Lab. Results		Heilman	A. S. H. V. E. Lab. Results	
		U	H	H	h_e	H_I
Horizontal	¾	2.84	0.788	0.765	1.68	0.244
Horizontal	1	2.72	0.934	0.924	2.52	0.271
Horizontal	1¼	2.63	1.13	1.16	1.74	0.305
Horizontal	4	2.36	2.78	2.79	....	....
Vertical	¾	2.71	0.749	....	1.88	0.240
Vertical	1	2.63	0.903	....	2.00	0.251
Vertical	1¼	2.55	1.10	....	2.00	0.306

U = Btu per hour per square foot per degree Fahrenheit steam to air; for bare pipe.
H = Btu per hour per foot length per degree Fahrenheit steam to air; for bare pipe.
h_e = Btu per hour per square foot per degree Fahrenheit insulated surface to air; for covered pipe.
H_I = Btu per hour per foot length per degree Fahrenheit steam to air; for covered pipe.

Table 2. Heat Emission for Bare Copper Pipe, Based on A. S. H. V. E. Laboratory Results

Position	Size (Inches)	U	H
Horizontal	¾	2.01	0.462
Horizontal	1	1.91	0.560
Horizontal	1¼	1.76	0.624
Horizontal	4	1.32	1.43
Vertical	¾	2.02	0.463
Vertical	1	1.92	0.574

U = Btu per hour per square foot per degree Fahrenheit steam to air; for bare pipe.
H = Btu per hour per foot length per degree Fahrenheit steam to air; for bare pipe.

Table 3. Heat Emission from One-Inch Horizontal Pipe with Various Surfaces in Btu per Hour per Square Foot of Surface per Degree Fahrenheit Difference Between Steam and Air

Type of Surface	Iron	Copper	Gal. Iron	Brass	Aluminum
Bare as received	2.72	1.91	1.78	1.71	1.93
Buffed	1.85	1.74	....	....	....
Black Duco	2.87	....	....	....	2.88
Lamp Black	2.88	....	....	....	....
Red Duco	....	2.89	....	....	....
White Duco	2.88	....	....	....	....
Aluminum Paint	2.24	....	....	....	....
Black Duco over Aluminum Paint	2.81	....	....	....	....
Aluminum Paint over Black Duco	....	....	....	...	2.19
Aluminum Paint over Black Duco over Aluminum Paint.	2.17	....	....	....	....

tion surface per degree temperature difference between the insulation surface and the air, and the heat emission in Btu per hour per foot length of insulated pipe per degree temperature difference between steam and air. Table 2 gives

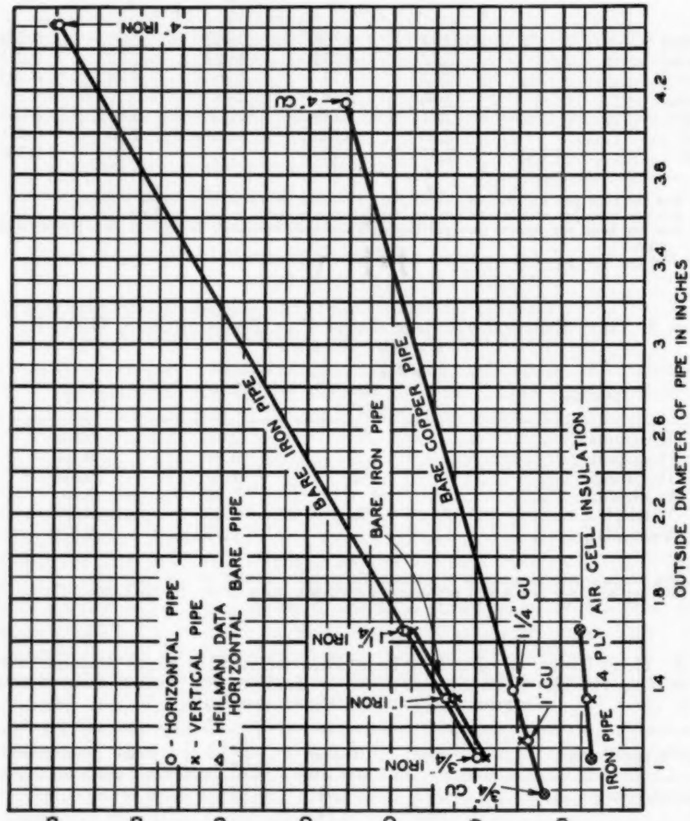


FIG. 4. RELATION BETWEEN HEAT EMISSION PER FOOT OF LENGTH AND DIAMETER OF PIPE

similar heat emission data for bare copper pipe in the vertical and horizontal positions. Table 3 gives the transmittance coefficients for bare, galvanized, aluminum and brass pipe, and for iron and copper pipe with other than *as received* surfaces. These results are for one-inch pipe only.

DISCUSSION OF RESULTS

The small temperature difference existing between the steam and the outside surface of the pipe bears out the generally accepted fact that the important

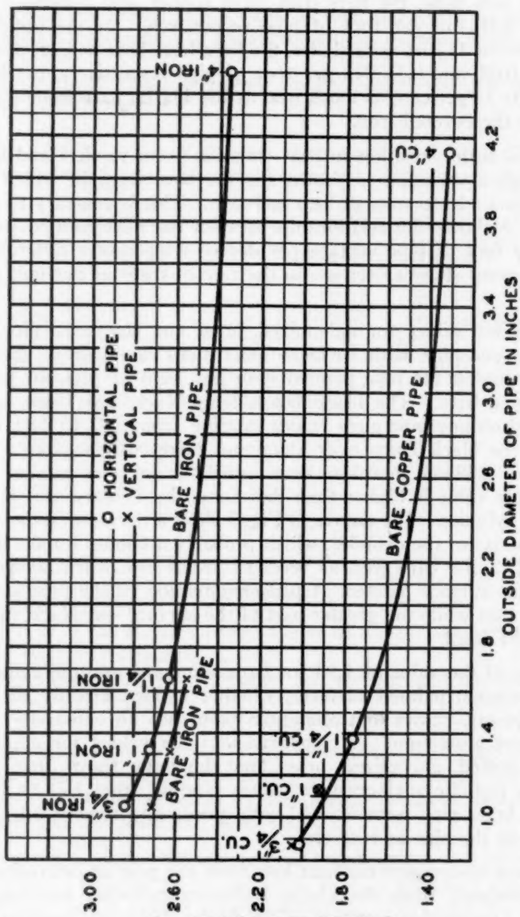


FIG. 5. RELATION BETWEEN HEAT EMISSION PER SQ. FT. OF SURFACE AND PIPE DIAMETER

factor in resistance to heat flow from a bare pipe is in the transfer from the metal surface to the air, rather than the thermal resistance of metal itself.

The heat transfer coefficients expressed in Btu per hour per foot length of

pipe per degree temperature difference steam to air, are plotted in Fig. 4 for bare iron, bare copper, and for insulated iron. When plotted against outside pipe diameter, substantially straight lines result. The bare iron curve fits almost perfectly the results of Heilman.[4]

For pipe of 1.65 in., outside diameter, which is the outside diameter of 1¼-in. normal size iron pipe, the bare iron, bare copper, and covered iron, emit 1.13, 0.70, and 0.31 Btu per foot of pipe respectively. If the comparison is made on the basis of 1¼-in. normal size, the bare iron, bare copper, and covered iron emit 1.13, 0.62, and 0.31 Btu per foot of pipe respectively, or the 1¼-in. copper tube emits 54 per cent as much heat as the 1¼-in. bare iron, and 200 per cent as much as the covered iron.

The data show little variation in heat emission from vertical and horizontal pipe. The greatest variation is shown for the bare iron, in which case the vertical pipe shows a lower rate of heat emission. This is contrary to what was expected. Fig. 5 shows the relationship between the heat emitted in Btu per hour per square foot of pipe surface per degree temperature difference steam to air. These curves show a decrease in the rate of transfer per unit area with pipe size.

The heat transfer values for galvanized, brass and aluminum pipe, and for the iron and copper pipe with surfaces other than *as received,* show higher rates of heat emission for pipe painted with any kind of pigment paint, than for the same pipe bare. The black Duco, lamp black, red Duco, and white Duco when used on any pipe gave values ranging from 2.87 to 2.89, excepting in the case of the black Duco over aluminum paint, which gives a value of 2.81. A value of 2.88 can therefore be accepted for surfaces painted with pigment paint. This value is higher than that found by Willard and Kratz[5] for a 10-in. painted cylinder. The curves in Fig. 5 show a decrease in the coefficient with increase in size of the cylinder, which probably accounts for the difference. The value of 2.72 for bare iron as received seems low when compared with 2.88 for the same surface painted. Upon examination of the pipe as received, it was observed that while the greater part of the surface was black, many small areas were bright metal.

A comparison of the value of 1.93 for bare aluminum and the values of 2.17 to 2.24 for aluminum painted surfaces, whether applied directly to the metal or over other paints, shows that aluminum paint has an emissivity not much greater than bare aluminum. A comparison of the values for copper as received, copper buffed, galvanized, brass and aluminum shows that 1-in. pipe of any of these metallic surfaces give values ranging from 1.71 to 1.93. The galvanized and brass pipe were washed with gasoline to remove oil and grease deposits found on the pipe as received.

It is of interest to compare the heat loss from the pipe as determined by test with values calculated from the Stefan-Boltzman radiation equation and the Rice[6]-Heilman[7] convection equation. The Stefan-Boltzman equation is:

[4] Heat Transmission from Bare and Insulated Pipes, by R. H. Heilman (*Industrial and Engineering Chemistry,* May, 1924).

[5] Heat Emission from the Surfaces of Cast Iron and Copper Cylinders Heated with Low Pressure Steam, by A. C. Willard and A. P. Kratz, A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931.

[6] Chester W. Rice (*A.I.E.E.* Transactions, Vol. 43, 1924).

[7] Surface Heat Transmission, by R. H. Heilman (*Mechanical Engineering,* May, 1929, Section 1).

$$H_r = 0.1723\, e\left[\left(\frac{T_1}{100}\right)^4 - \left(\frac{T_2}{100}\right)^4\right]$$

where

H_r = heat loss by radiation, Btu per square foot per hour
e = emissivity coefficient
T_1 = absolute temperature of the hot surface, degrees Fahrenheit
T_2 = absolute temperature of the surroundings, degrees Fahrenheit

Values for e which might be chosen for the copper and iron surfaces studied are given in Table 4.

TABLE 4. VALUE OF e FOR COPPER AND IRON SURFACES

Surface	e	Authority
Wrought Iron high polish	0.28	Marks Handbook [a]
Cast Iron—bright	0.21	International Critical Tables
Rough Steel	0.94	Heilman [b]
Copper—buffed high polish	0.10	Marks Handbook
Copper—buffed	0.167	Hütte [c]
Copper—oxidized	0.57	International Critical Tables

[a] Transmission of Heat by Radiation, Revised by H. C. Hottel. (Marks Handbook.)
[b] Surface Heat Transmission, by R. H. Heilman. (*Mechanical Engineering*, Section 1, May, 1929.)
[c] Des Ingenieurs Taschenbuch, Twenty-second edition.

The Rice-Heilman equation is:

$$H_c = C\left(\frac{1}{D}\right)^{0.2} \times \left(\frac{1}{T_a}\right)^{0.181} \times \Delta t^{1.266}$$

Where

H_c = heat loss by convection, Btu per square foot per hour
C = a constant depending on configuration. 1.016 for horizontal cylinders.
D = diameter, inches
T_a = average of the absolute temperatures of the surface and the surrounding air, degrees Fahrenheit
Δt = temperature difference between the surface and the surrounding air, degrees Fahrenheit

Table 5 gives the heat loss in Btu per hour per square foot per degree temperature difference for copper and iron pipe of different surfaces as found by the Laboratory and as calculated by the Stefan-Boltzman equation, the Rice-Heilman equation, and assumed values for e.

The calculated value for the copper pipe based upon $e = 0.167$ gives 1.42 Btu compared with 1.74 found by the A. S. H. V. E. Laboratory for a buffed pipe, and 1.91 found for the copper pipe as received. Obviously the value $e = 0.10$ for a highly polished copper and the value of 0.57 for oxidized copper are too low and too high respectively for the buffed and as received copper pipe studied by the A. S. H. V. E. Laboratory.

The values of $e = 0.21$ and 0.28 for bright cast iron and polished wrought iron giving calculated values of 1.45 and 1.56 are too low for the buffed iron pipe

TABLE 5. HEAT LOSS FROM ONE-INCH LOW PRESSURE STEAM PIPE AS FOUND BY TEST AND BY CALCULATION

Kind of Pipe	Assumed Value of e	H_r	H_c	$\frac{H_r+H_c}{\Delta t}$	A.S.H.V.E. Laboratory
Copper as received	0.57	122.97	166.12	2.03	1.91
Copper—buffed	0.167	35.60	165.90	1.42	1.74
Copper—buffed	0.10	22.81	165.90	1.29	1.74
Iron—black	0.94	198.67	154.06	2.56	2.72
Iron—buffed	0.21	45.43	162.03	1.45	1.85
Iron—buffed	0.28	60.57	162.03	1.56	1.85
Any pipe covered with pigment paint	0.94	198.67	154.06	2.56	2.87 to 2.89

H_r = Heat emission by radiation in Btu per hour per square foot of surface for the existing temperature difference.

H_c = Heat emission by convection in Btu per hour per square foot of surface for the existing temperature difference.

$\frac{H_r + H_c}{\Delta t}$ = Total heat emission in Btu per hour per square foot of surface per degree temperature difference.

which Laboratory test found to have an emission of 1.85 Btu per hour per square foot per degree temperature difference. The value of $e = 0.94$ giving a calculated coefficient of 2.56 is too low for the black iron, which by test gave a coefficient of 2.72, and is also low for the pigment painted surfaces tested by the Laboratory, which gave coefficients ranging from 2.87 to 2.89 Btu per hour per square foot per degree temperature difference.

These comparisons show that with the selection of the proper value for e the rate of heat emission for any case in hand may be accurately computed. However, the selection of the value of e for any particular case is often difficult.

VALUES OF e FOR VARIOUS SURFACES [a]

Surface	Temperature, Degrees Fahrenheit				
	100	200	300	400	500
Polished silver	0.0221	0.0252	0.0292	0.0315	0.0295
Lampblack	0.945	0.945	0.945	0.945	0.945
Asbestos Paper	0.930	0.934	0.943	0.955	0.929
Rough Steel Plate	0.945	0.950	0.955	0.961	0.969
Aluminum-Surfaced Roofing	0.216	....	....	....	....
Polished Brass	0.096	0.096	0.098	0.098	0.096
Flat Black Lacquer	0.96	0.98	....	....	....
Black Lacquer	0.80	0.95	....	....	....
White Lacquer	0.80	0.95	....	....	....

[a] A. S. H. V. E. GUIDE 1931.

SUMMARY AND CONCLUSIONS

1. The heat loss from bare copper pipe is approximately 54 per cent of the loss from bare black iron pipe of the same nominal size, and 203 per cent of the loss from iron pipe covered with four-ply air cell insulation.

2. The heat loss is approximately the same for horizontal and vertical pipe of the same size and material.

3. The heat loss from pipe may be calculated from the Stefan-Boltzman and the Rice-Heilman equations with a high degree of accuracy by using the proper shape factor and value of *e*. However, the choice of *e* for the case in hand is important.

DISCUSSION

C. A. HILL (WRITTEN): The engineer who uses copper pipe as well as the makers should feel equally pleased to have the definite information given in this paper.

There has long been a general recognition of the value of brass or copper pipe for hot water lines due to the freedom from corrosion and rusting. There are sections of the country where water conditions made the use of such materials necessary and indeed some where copper has been used for years as the most satisfactory material for hot water.

The reduced cost of piping systems using the new sizes and weights of copper with the fittings designed for them has caused much of this material to be used, not only on hot water but on low pressure steam lines. While some 8-in. and much 6-in. pipe has been used, most of it has been the smaller sizes. It has been used for mains, risers and branches as well as for returns.

Many of the larger lines and nearly all the smaller ones have been left uncovered due to the correct assumption that the heat losses would be but a fraction of bare iron pipe losses.

Definite figures were needed and as a result of this investigation the engineer can decide when the considerations of space, cost, beauty of appearance and freedom from corrosion will justify the use of copper pipe on steam lines. One thing is certain, he can use copper pipe on all lines carrying any hot water with the assurance that he will not have to make any allowance for their ultimate plugging by rust.

Where excessive fuel costs are a factor or places where any heat loss would be objectionable the pipe can be covered but when each individual job is considered in the light of the facts brought out by this paper more copper will be used and most of it will be used uncovered.

F. W. HVOSLEF: Why was the copper pipe "specified as received"? Does that mean a bright new shiny copper pipe or copper after it had been lying in the atmosphere for a while?

F. C. HOUGHTEN: The values referred to are for copper as purchased in the market, and not copper that had been buffed. Buffing the copper brought the heat emission down considerably.

MR. HVOSLEF: Within the past month I have been approached by a salesman of copper tubing and fittings who tried to persuade me that I should use copper for steam mains because I would not have to cover them, basing his statement on this paper.

W. J. KING: In reply to the last comment if a considerable thickness of poor insulation is used with small pipes the amount of surface exposed can be increased sufficiently to more than offset the effect of the insulation blanket. Therefore, it is an advantage to use the copper pipe having a lower surface emissivity. It is better to use a bare pipe than a poor insulation. Matters can be improved by using a high grade insulation.

Results of other investigators on this same general problem in Europe have borne out these developments. Practically the same coefficients have been found for the horizontal or vertical position.

For small pipes the curves in Fig. 5 showing the effect of diameter cannot be extrapolated without considerable error. These coefficients might be increased many times. The curves go up steeply for the smaller sizes.

T. B. SMITH: What effect would the oxidation of the copper have on the heat emission of the copper pipe?

F. C. HOUGHTEN: Some idea of the rate of emission may be obtained by referring to Table 4 which gives the generally accepted emissivities for the different surfaces found in handbooks. Emissivities range from 0.10 to 0.57 for copper, depending on whether it is highly polished, buffed or oxidized.

C. A. HILL: Concerning copper surfaces it should be noted that the process of annealing makes a surface change which, while it is not oxidized, under a microscope it is considerably rougher than the hard copper. In fact, the hard copper has a brightly burnished surface and the same material when annealed is relatively rough and doubtless would make a considerable difference in this emission coefficient.

CHAIRMAN ROWLEY: Are there any further remarks? In closing I might say that it is evident that changing the surface of the material is the effect which actually changes the coefficient and the conductivity.

No. 919

SUPPLEMENTARY FRICTION HEADS IN ONE-INCH CAST-IRON TEES

By F. E. GIESECKE [1] (*MEMBER*) AND W. H. BADGETT [2] (*NON-MEMBER*)
COLLEGE STATION, TEXAS

This paper is the result of research conducted at Agricultural and Mechanical College of Texas in cooperation with the A. S. H. V. E. Research Laboratory

IN THE first report of this study, which was presented at the Semi-Annual Meeting of the Society, Swampscott, Mass., June, 1931,[3] it was shown that the loss of head in a tee varies with the direction in which the water flows through the tee and also with the proportion of the water which is diverted at right angles in the tee.

It was shown, for example (see Figs. 1 and 2) that when the water enters at one end of the tee and 50 per cent is diverted while 50 per cent flows straight through the tee, the loss of head for the diverted portion is equal to that in four elbows and the loss of head for the remaining 50 per cent, which flows straight through the tee, is equal to that in 0.6 elbows. In expressing the loss of head in a tee in terms of the loss of head in an elbow, *i.e.*, in elbow equivalents, the loss of head in one elbow was assumed equal to $0.9\frac{v^2}{2g}$, because in an earlier investigation, the loss of head in a standard 1-in. elbow had been found equal to this quantity. In more recent investigations by A. P. Kratz, H. J. Macintire, and R. E. Gould at the University of Illinois,[4] it was found that for 1¼-in. and 2-in. elbows the losses of head varied from $0.89\frac{v^2}{2g}$ to $1.33\frac{v^2}{2g}$, aver-

[1] Director, Texas Engineering Experiment Station, Agricultural and Mechanical College of Texas, College Station, Texas.
[2] Research Assistant, Texas Engineering Experiment Station, Agricultural and Mechanical College of Texas, College Station, Texas.
[3] See Friction Heads in One-Inch Standard Cast-Iron Tees, by F. E. Giesecke and W. H. Badgett, A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931.
[4] See University of Illinois Engineering Experiment Station *Bulletin* No. 222, pp. 21 and 22.

Presented at the 38th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Cleveland, Ohio, January, 1932.

aging $1.035\frac{v^2}{2g}$ for 104 tests. It will therefore be sufficiently accurate for practical purposes to place the loss of head in one elbow equal to $\frac{v^2}{2g}$. This will be done in the following discussions.

LOSS OF HEAD FOR STRAIGHT FLOW

Before the tests of the losses of head in tees had been performed, it was believed that the loss of head for the portion of water flowing straight through the tee would be practically zero and might even be negative because when a

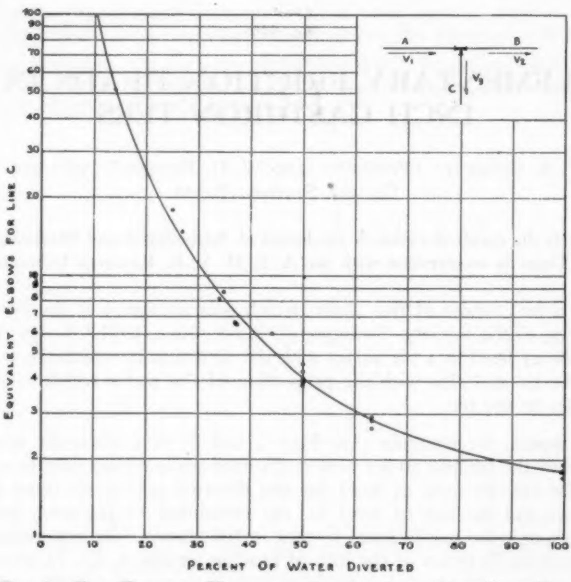


FIG. 1. THE FRICTION HEAD IN A 1 x 1 x 1-IN. CAST-IRON TEE WHEN WATER ENTERS AT *A* AND A PORTION IS DISCHARGED AT *C*, IN TERMS OF THE FRICTION HEAD OF AN ELBOW AT *C*

portion of the water is diverted the velocity of the remainder is reduced and it was believed that the resulting reduction of velocity head would be as large and possibly larger than the friction head in the tee.

Since the tests proved conclusively that this was not the case, it was decided to substitute a glass tee for the cast-iron tee and to observe the flow of water through the glass tee. In order to make the stream lines visible, small bubbles of gas (hydrogen and oxygen) were produced electrolytically in the fluid stream by a method similar to that described by J. Zenneck.[5] The resulting streams of

[5] Transactions, *German Association of Physicists*, Vol. XVI, No. 14.

gas bubbles could be seen very clearly and could be photographed when they were illuminated by a thin sheet of light at right angles to the direction from which the photograph was taken. Fig. 3 shows the stream lines and eddies in the glass tee when about 50 per cent of the water was being diverted. It is evident from the picture that no portion of the water flows straight through the tee, but that the entire current is deflected to such an extent that considerable eddying occurs also in the straight portion of the tee. This eddying is, no doubt, the cause of the unexpected loss of head in that portion of the current

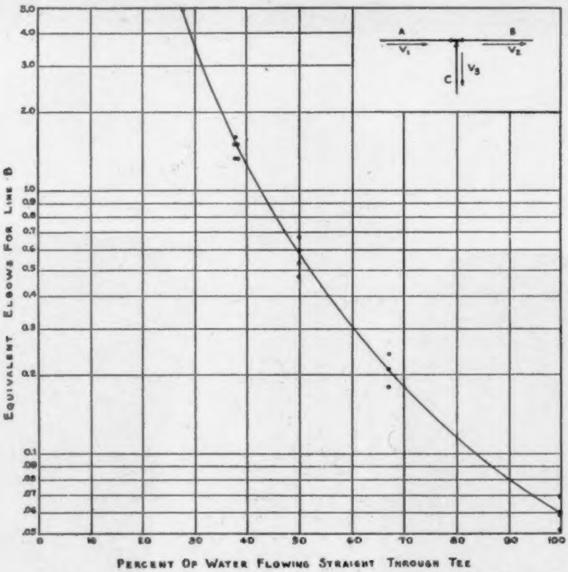


FIG. 2. THE FRICTION HEAD IN A 1 x 1 x 1-IN. CAST-IRON TEE WHEN WATER ENTERS AT *A* AND A PORTION IS DISCHARGED AT *B*, IN TERMS OF THE FRICTION HEAD OF AN ELBOW AT *B*

which passes straight through the tee and which was disclosed by the recent tests.

TESTS FOR VARIATIONS IN FLOW DIRECTION

Having determined the losses of head when the water enters the tee at *A* and is discharged at *B* and *C* (Figs. 1 and 2), additional tests were made to determine the losses of head when the water enters at *A* and *C* and is discharged at *B* (Figs. 4 and 5); when it enters at *C* and is discharged at *A* and *B* (Fig. 6); and when it enters at *A* and *B* and is discharged at *C* (Fig. 7). The diagrams of these figures are to be used as follows:

1. In Fig. 4, if 25 per cent of the water flowing through the tee enters from

C with the velocity v_3, the loss of head in the tee, for the path from C to B is equal to 10 elbow equivalents of the line C, i.e., equal to $10\frac{v_3^2}{2g}$.

2. In Fig. 5, if 75 per cent of the water flowing through the tee enters from A with the velocity v_1, the loss of head in the tee for the path AB is equal to

FIG. 3. THE STREAM LINES AND EDDIES OF WATER FLOWING THROUGH A TEE WHEN ABOUT 50 PER CENT OF THE WATER IS DIVERTED

1.25 elbow equivalents of the line A, i.e., equal to $1.25\frac{v_1^2}{2g}$. It is interesting to note how much larger the loss of head is in the case of Fig. 4 than in the case of Fig. 2 for the water flowing straight through the tee; with 50 per cent flowing straight through, the two losses of head are 2.25 and 0.6 elbow equivalents, respectively.

3. In Fig. 6, if 25 per cent of the water flowing through the tee is discharged

at A (or B) the loss of head in the tee for the path CA (or CB) is equal to 11 elbow equivalents for the line A (or B), i.e., equal to $11\frac{v_1^2}{2g}$ (or $11\frac{v_2^2}{2g}$).

4. In Fig. 7, if 25 per cent of the water flowing through the tee enters at A (or B) the loss of head in the tee for the path AC (or BC) is equal to 16 elbow equivalents for the line A (or B), i.e., equal to $16\frac{v_1^2}{2g}$ (or $16\frac{v_2^2}{2g}$).

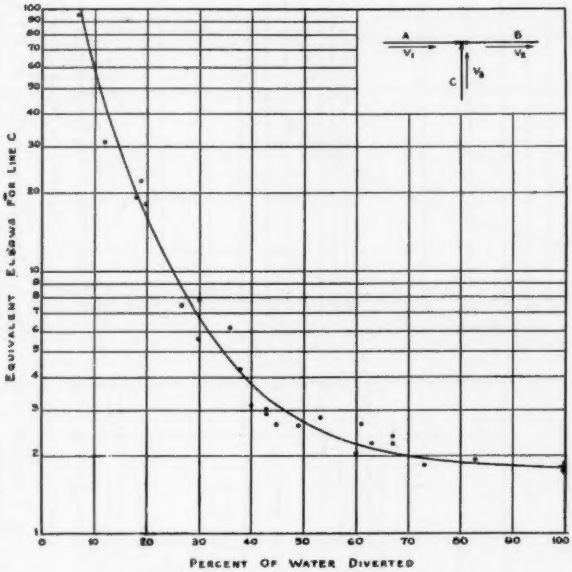


Fig. 4. The Friction Head in a 1 x 1 x 1-in. Cast-Iron Tee When Water Enters at A and C and Is Discharged at B, in Terms of the Friction Head of an Elbow at C

The diagrams of Figs. 1, 5, 6 and 7 are very similar. They are shown collectively by the four dashed light lines in Fig. 8. The differences between these four curves are so slight that they may be neglected for practical purposes and the four curves replaced by one curve as shown by the heavy line of Fig. 8.

It should be noted that as the water, when diverted to C (Fig. 1), approaches the tee from A with the velocity v_1, the velocity in the direction AB is changed from v_1 to zero, and that the corresponding velocity head $\left(\frac{v_1^2}{2g}\right)$ must be transformed partly into heat and partly into pressure head; also, as the water flows from the tee to C its velocity increases from zero to v_3 and that the

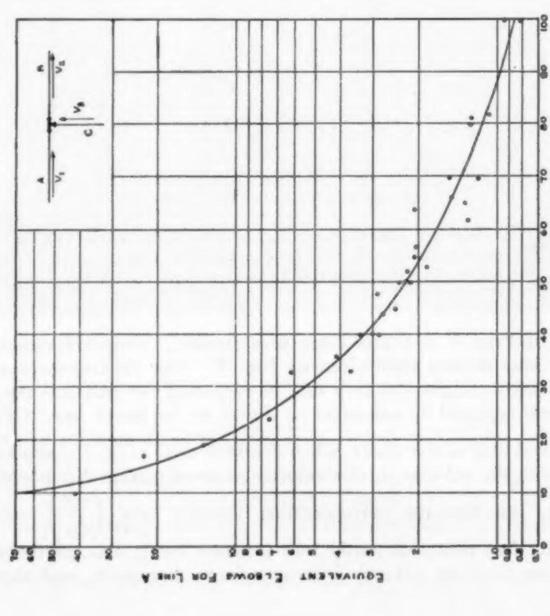


FIG. 5. THE FRICTION HEAD IN A 1 x 1 x 1-IN. CAST-IRON TEE WHEN WATER ENTERS AT A AND C AND IS DISCHARGED AT B, IN TERMS OF THE FRICTION HEAD OF AN ELBOW AT A

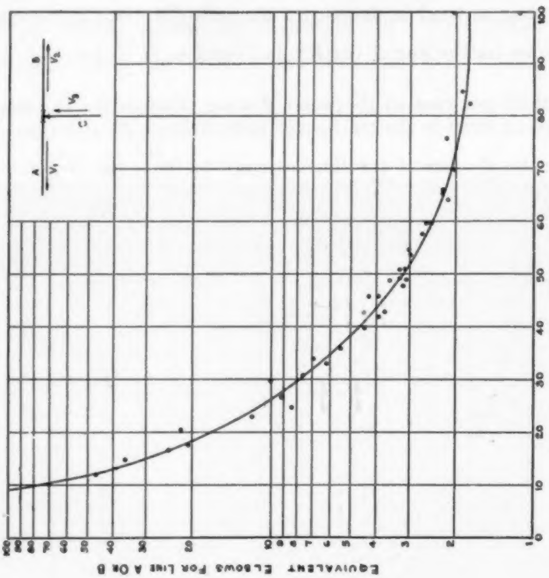


FIG. 6. THE FRICTION HEAD IN A 1 x 1 x 1-IN. CAST-IRON TEE WHEN WATER ENTERS AT C AND IS DISCHARGED AT A AND B, IN TERMS OF THE FRICTION HEAD OF AN ELBOW AT A OR B

corresponding velocity head $\left(\frac{v_3^2}{2g}\right)$ must be produced by a corresponding reduction in the pressure head. During one of the tests an effort was made to determine how much the pressure head builds up in the tee as the velocity of the water, in the direction from A to the tee, is reduced from v_1 to zero, and it was found that the increase in pressure head was too small to be measured by the methods used in the test. If it is assumed that the entire velocity head $\left(\frac{v_1^2}{2g}\right)$ is transformed into heat by the eddy currents in and near the tee, and if

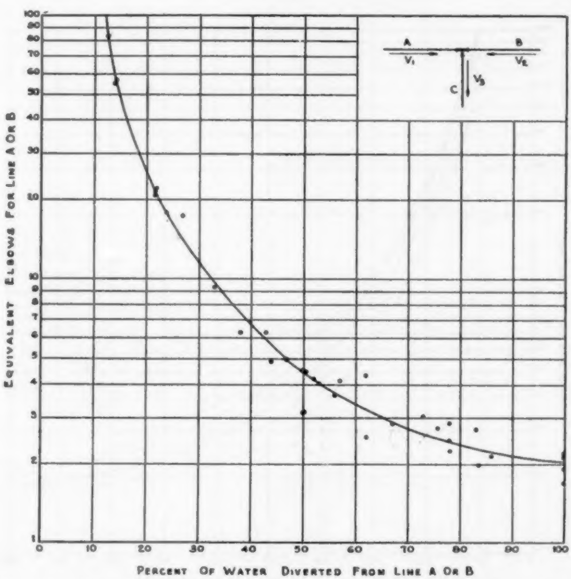


Fig. 7. The Friction Head in a 1 x 1 x 1-in. Cast-Iron Tee When Water Enters at A and B and Is Discharged at C, in Terms of the Friction Head of an Elbow at A or B

it is also assumed that the entire velocity head $\left(\frac{v_3^2}{2g}\right)$ is produced at the expense of the available pressure head, the total loss of head in the tee will be $\frac{v_1^2}{2g}$ plus $\frac{v_3^2}{2g}$.

Comparison of Calculated and Experimentally Determined Heads

To compare the losses of head calculated in this manner with those found experimentally and shown in Figs. 1, 5, 6 and 7, v_1 was expressed in terms of v_3 as follows:

If 50 per cent of the water is diverted, v_1 is equal to $2v_3$ and $\frac{v_1^2}{2g}$ is equal to $4\frac{v_3^2}{2g}$, and the total loss of head is equal to $5\frac{v_3^2}{2g}$ or to 5 elbow equivalents.

Similar calculations were made for other percentages and the results shown in Fig. 8 by small circles connected by a solid line. It will be noticed that this line practically coincides with the line shown in Fig. 7 for the case when the two currents from A and B impinge upon each other, but that it gives values

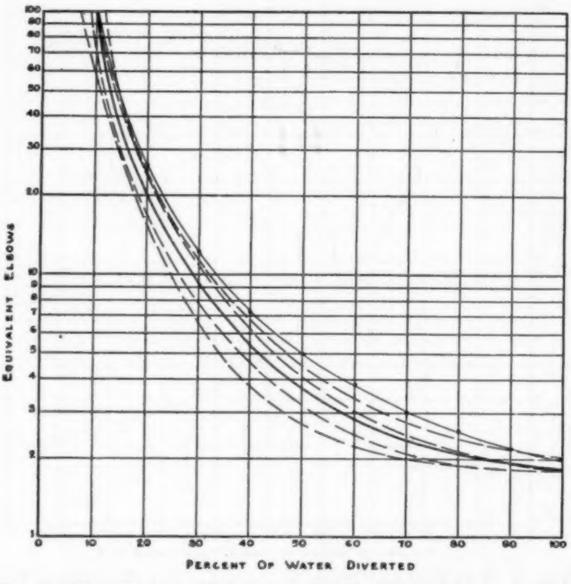


FIG. 8. THE FRICTION HEADS IN A 1 x 1 x 1-IN. CAST-IRON TEE AS SHOWN IN FIGS. 1, 5, 6 AND 7, WITH AN AVERAGE OF THESE FOUR LINES, AND A LINE SHOWING THEORETICAL VALUES CALCULATED BY AN APPROXIMATE METHOD

about 20 per cent higher than the average of the four curves determined experimentally.

It seems from this comparison that the suggested method of calculating the losses of head in tees may be used with safety for all cases. For example, for a 1½ x 1¼ x 1-in. tee, placed as shown in Fig. 1, assume that 25 per cent of the water is diverted into the 1-in. pipe; v_1 will then be equal to v_3 multiplied by 4, and divided by 2.36, the ratio of the two pipe areas; *i.e.*, v_1 will be equal to *1.7* v_3. The corresponding loss of head in the tee will be $\frac{v_1^2}{2g}$ plus $\frac{v_3^2}{2g}$ or

$3.9\frac{v_3^2}{2g}$, or 3.9 elbow equivalents for the line C.

For the losses of head in the currents passing straight through the tee, in the cases shown in Figs. 2 and 5, it is evident that the losses in the first case are so small that they may generally be neglected, but in the second case they are of sufficient magnitude to be considered in accurate calculations especially when the portion flowing straight through is from 25 to 75 per cent of the total.

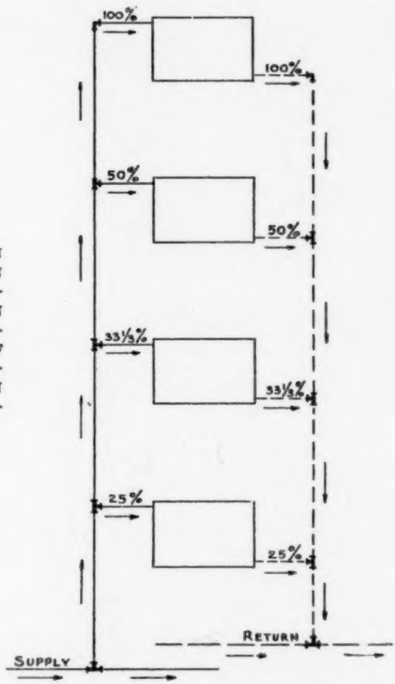


Fig. 9. A Common Installation in Which the Friction Heads in Tees Are an Important Part of the Total Friction Heads in the Several Circuits

Application of Results to Typical Case

A case which occurs frequently in practice and for which it is important to know the losses of head in tees is shown in Fig. 9. When the four radiators are of equal size, one-fourth of the water is diverted to the first floor radiator, one-third to the second floor radiator, and one-half to the third floor radiator. If the installation shown were a part of a forced circulation system using 1-in. pipe for the two risers and the four radiator connections, in an extreme case, the losses of head in the three pairs of tees would be, according to Figs. 1 and 4, for the first, second and third floor radiators, 26, 15 and 7 elbow equivalents

respectively. As a result, the first floor radiator would be at a considerable disadvantage when compared with the second or third floor radiators. This disadvantage, resulting from lack of knowledge regarding the losses of head in tees, may be responsible in part for the frequently unsatisfactory operation of first floor radiators when connected as shown in Fig. 9.

No. 920

SOME FUNDAMENTAL CONSIDERATIONS OF CORROSION IN STEAM AND CONDENSATE LINES

By R. E. HALL,[1] PITTSBURGH, PA., AND A. R. MUMFORD,[2] NEW YORK, N. Y.
NON-MEMBERS

THIS paper presents the results of an extensive investigation, covering a period of one year, into the factors influencing corrosion in steam systems. The investigation includes a study of the raw water and the production, the distribution, and the utilization of steam, with reference to heating systems and appliances only. Data are presented in full showing the extent and constitution of deposits in the heating systems of a large office building and a large hotel in New York City. These deposits were found to originate from the action of corrosion and were found to contain little or no evidence of carryover from the boilers. The cause of the excessive corrosion was found to be the inleakage of air into the vacuum return system. The amounts of oxygen and carbon dioxide associated with the steam are shown to be of insignificant importance as corrosive agents by the application of the Law of Henry and Dalton. The cause of corrosion trouble in heating systems can, therefore, be sought in the operation and, to a small extent, in the design of the systems and not in the quality of steam used if that quality is as good as that used in New York City.

THE SOLVENT ACTION OF WATER IN THE STEAM HEATING CYCLE

Water is universally a solvent of the substances with which it contacts. Some substances, such as sugar, dissolve readily in water. Others, such as limestone, dissolve far less readily but nevertheless are dissolved by water. Since one of the purposes of this paper is to specify available means of defense against deleterious effects of the solvent action of water, attention will be directed first to certain laws of solubility and to certain mechanisms of reactions, a knowl-

[1] Director, Hall Laboratories, Pittsburgh, Pa.
[2] Research and Design Engineer, New York Steam Corporation.
Presented at the 38th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Cleveland, Ohio, January, 1932, by A. R. Mumford.

edge of which is requisite for the sagacious construction of adequate defense against these effects.

A compound substance may dissolve without chemical reaction with the water or with chemical reaction. Neglecting such chemical actions as are characterized by the acquisition or loss of so-called water of crystallization, most compounds dissolve in water without chemical reaction. The process of dissolution in such cases is characterized by a physical division of the crystals into molecules which disperse themselves through the water. Such compounds are further subdivided in solution by the breaking up of molecules into parts called ions. This process is called dissociation and each compound dissociates to a definite degree which is specific for that compound and which varies from a value of zero to one of almost one hundred per cent for different compounds in saturated solutions. The degree of dissociation for a particular compound is called the dissociation constant and this numerical value is the product of the equivalent concentrations of the two or more ions formed. If a substance enters into chemical reaction with water (other than actions involving water of crystallization) the reaction is between ions and the solution of the substance involves chemical changes which cannot be reversed under practical conditions whereas the physical changes in dissolution of the first class can be reversed by evaporating the water.

Solution of Ordinary Crystalline Compounds

When a small amount of sugar or salt is added to water the sugar or salt disappears more or less rapidly depending upon the amount of stirring. If the addition of sugar or salt is continued the water eventually becomes saturated and in this condition no further visible solution of the material will take place. However, although the total quantity of sugar or salt molecules in solution has become constant, the personnel is constantly changing because some molecules are continuously dissolving and an equal number of others are just as rapidly returning to the visible crystals.

The quantity of any substance which will dissolve in a given quantity of water is a measure of the specific solubility of the substance. The specific solubility of each substance is not only different from that of all other substances but the solubility of each changes in a different degree with the temperature of the water. Some substances become more soluble as the temperature rises while others become less soluble under like conditions. The action of each individual substance, when placed in contact with water, must be studied to learn the properties of the substance and no predictions of solubility can be made without such studies.

Dissociation of Water

Pure water contains not only molecules of water (H_2O) but separated parts of these molecules called ions (H^+, and OH^-). In pure water the amounts of hydrogen ion (H^+) and hydroxyl ion (OH^-) are equally small and, as was the case in a saturated solution, new molecules are continually splitting up at a rate equal to that at which hydrogen ions are combining with hydroxyl ions. This condition of equilibrium is represented by the equation,

$$\text{Water} \rightleftharpoons \text{Hydrogen ion} + \text{Hydroxyl ion.}$$

Under conditions of equilibrium the product of the equivalent concentrations

of hydrogen-ion and hydroxyl-ion is a constant. This constant is represented by the following equation for conditions at room temperature.

$$(H^+) \times (OH^-) = 1 \times 10^{-14}$$

The importance of this dissociation constant cannot be over-estimated in any study of corrosion. Any increase in the concentration of either ion must cause a decrease in the concentration of the other ion if the product is to remain constant. Consequently, if, for any reason, it is desirable to reduce the concentration of hydrogen ion such reduction can be readily accomplished by increasing the concentration of hydroxyl ion.

For convenience a number representing the logarithm of the reciprocal of the hydrogen-ion concentration is used to define the relation of (H^+) and (OH^-) in any solution. This number is called the *pH* value. A *pH* value greater than 7 means that the solution contains more (OH^-) than (H^+) and is therefore alkaline, whereas a *pH* value less than 7 means that the solution contains more (H^+) than (OH^-) and is therefore acid.

Solution of Metals

Because iron is the material of which most pipe is made, the mechanism of its solution in water will be discussed first. When iron is placed in water, it replaces some of the hydrogen ion, and goes into solution. In this process the hydrogen-ion concentration is reduced and more water ionizes to make up the deficiency. Because for each additional molecule of water which becomes ionized an additional hydroxyl is formed, the hydroxyl-ion concentration is continuously increasing during this process and because the product of the concentrations of (H^+) and (OH^-) must be a constant, a point is reached at which the concentration of hydrogen ion is so small that iron is no longer able to replace it and the solvent action of water on iron ceases. The chemical change involved in the solution of iron has, for one of its products ferrous hydroxide $(Fe(OH)_2)$ a soluble salt of iron. This compound ionizes to such an extent that the *pH* value of a solution of ferrous hydroxide is 9.6 and at this *pH* value an effect similar to *back pressure* prevents the formation of more ionic iron and thus prevents the solution of metallic iron. Whether ferrous salt is in solution or not, if the *pH* value is 9.6, because of the presence of sufficient hydroxyl ion, metallic iron cannot dissolve.

If, instead of permitting the dissolution of iron to build up the hydroxyl-ion concentration by using up the hydrogen ion, we increase the concentration of hydroxyl ion by adding caustic soda until the *pH* value of the solution is 9.6, we attain the same result, namely, prevention of the solution of iron.

If, on the other hand, we increase the concentration of hydrogen ion by adding sulphuric acid, and thus replenish the supply perhaps many times faster than the solution of iron would normally use it up, we increase both the rate of reaction and the quantity of iron which dissolves. Solution of the iron will continue until the concentration of the hydrogen is so small that the iron is no longer able to effect replacement.

The solvent action of water on a metal is a function of the hydrogen ion concentration of the water, and the power of the metal to replace hydrogen ion. The power of each metal to replace hydrogen ion in water differs from that of all other metals and the metals are usually listed in the order of the

power. Of the common metals, copper, nickel, tin and lead have less power and aluminum and zinc have a greater power than iron to replace the hydrogen ion in water. The rapidity with which each metal replaces the hydrogen ion in water can be varied by placing in contact with the metal another metal with a more or less widely different power. The net difference in power, of the two contacting metals, expressed in volts is the force leading to the dissolution of the metal with the greater power and the arrangement is an electrolytic cell.

If instead of a different metal, an alloy of the principal metal is used the same action, although to a lesser degree, takes place. Such alloys or impurities are introduced in manufacturing but, during the past years, manufacturers have successfully reduced the danger of failure ascribable to the local segregations or inclusions of foreign material in their product by adjusting the process to assume uniformity of composition.

Mechanical strain introduced by working the metal is sufficient to set up an electrolytic cell at such points as nipple threads and bends and thus accelerate the normal rate of corrosion. The basic reaction is the same in all cases and the inhomogenity or strain is superimposed as an accelerant.

The apparent contradiction between the excess power of aluminum and zinc to replace hydrogen ion and the use of these metals in exposed conditions in preference to iron is answered by the presence of protective films. Both zinc and aluminum are protected by a thin film of oxide which segregates the metal from any contact with water and this film, if broken by a scratch or otherwise, is usually rebuilt by the corrosion process itself. If such a protective film, which constitutes the main defense against all corrosion of all types, is not rebuilt, corrosion occurs.

Iron has not the property of building a protective impermeable film on its exposed surface, except possibly in a solution having the correct *pH* value. A film of iron oxide is permeable both to gas and water and consequently offers no protection to the metal underneath. Iron in any form is the same and its power to replace hydrogen ion from water is nearly enough equal in all forms so that any differences in rates of dissolution ascribable to its form are of minor and not major importance. Unless aided by the establishment and maintenance of the requisite *pH* value in the contacting water no one of the forms is more or less able to protect itself by the formation of an impermeable protective coating. If, under like circumstances, real differences in the rates of corrosion of the different forms occur, it is due to other factors than the metal. Thickness of metal, in one form, results in a longer life under some conditions and adventitious protective coverings or inclusions may increase the service life. The effect of such coverings is excellent if they are everywhere interposed between the metal and the water but they are not rebuilt by the corrosive action and once the water passes them corrosion will proceed until the next covering layer is reached. Because such protection is adventitious and not of orderly planning its effectiveness can only be estimated by the average of many obser-vtions and does not lend itself to prediction for any specific case. We can conclude, therefore, that the service conditions are the dominant factors in determining the service life of iron or steel and that differences of internal conditions are merely subsidiary to the main factors.

Of the common metals, lead, zinc, tin, nickel, aluminum and chromium have the property of interposing an impermeable layer of the metallic oxide between

the metal and water. This layer is self-built and constitutes the principal reason for the resistance to corrosion. If such metals are successfully alloyed with iron the film forming property is retained in the alloy and, if the content of the film rebuilding metals is sufficient, the iron itself is protected from the solvent action of water. The film forming property is of major importance and overshadows such differences as may exist in power to replace the hydrogen ion in water. To alloy with iron a metal of less power to replace hydrogen ion, which metal does not have the film forming property, is usually not sufficient to make service conditions a minor factor in service life. Alloy steels which have properties of a nature to render them dominant over the service conditions to which they are subjected are not usually economical enough for pipe service.

The magnitude of the solvent action of pure water alone on metal is small, but other factors which we will consider, accelerate the solvent action, and multiply and magnify the deleterious results so that they become major considerations in the economical operation of the system.

It is necessary in any study of the resistance of various metals to corrosive action to emphasize service conditions as the major factor in operating life. The adventitious character of many of the differences in resistance to corrosion maces it very risky to predict performance unless the basis of the prediction is the history of service life in a large number of installations under like conditions.

GASES OF THE ATMOSPHERE: LAW OF HENRY AND DALTON

Oxygen

Of the many gases in the atmosphere we will consider only oxygen and carbon dioxide because the others are of negligible importance to the solvent action of water on iron.

Oxygen gas dissolves in water when molecules of the gas penetrate the surface of the liquid and finally become distributed throughout its depth. Temporarily those layers of water near the surface will contain more oxygen molecules per unit volume than lower layers but the motion of the molecules will gradually disperse them through the liquid volume and, indeed, some of them will escape from the liquid at its surface and reenter the gaseous phase. When equilibrium is established the number of molecules of oxygen leaving the surface of the water is the same as the number entering the water from the gaseous phase and the water is saturated.

Obviously the number of molecules of oxygen entering the water depends on the number of such molecules in the gaseous phase and therefore the number of molecules of oxygen in solution will depend on the same factor. A reduction in the number of molecules of oxygen in the gaseous phase may be obtained by reducing the total gas pressure or by diluting the oxygen with some other gas such as nitrogen keeping the total gas pressure constant.

Partial Pressure

If a gas is diluted with another gas and the total pressure is kept the same, that part of the pressure due to the presence of the original gas is reduced by the partial pressure of the diluent gas. Thus in the atmosphere the total pressure is about 15 lb per square inch and this pressure is made up principally of the

partial pressures of oxygen, nitrogen and carbon dioxide. The partial pressure of each constituent is in proportion to the ratio of the volume of each constituent to the total volume. In the case of the atmosphere, which is 21 per cent oxygen, 0.03 per cent carbon dioxide and the rest several inert gases, the partial pressure of oxygen is 3.15 lb per square inch and of carbon dioxide 0.045 lb per square inch.

Law of Henry and Dalton

From these considerations, it follows that the concentration of a gas such as oxygen dissolved in a liquid such as water is proportional to its partial pressure

TABLE 1. SOLUBILITY OF THE OXYGEN OF THE AIR IN WATER
When Partial Pressure of Water Vapor + Partial Pressure of Air = 14.696 *lb*

Temperature (Deg Fahr)	Vapor Pressure of Water (lb per sq in.)	Partial Pressure of Air (lb per sq in.)	Solubility	
			Pure Oxygen ml./l (32 F, 14.696 lb per sq in.)	Oxygen of Air 21% O_2 ml./l (32 F, 14.696 lb per sq in.)
32	0.0887	14.607	48.89	10.20
40	0.1217	14.574	43.50	9.05
50	0.1780	14.518	38.02	7.9
60	0.2561	14.440	33.8	7.0
70	0.3628	14.333	30.5	6.2
80	0.5067	14.189	27.5	5.6
90	0.6980	13.998	25.4	5.1
100	0.9487	13.747	23.1	4.65
110	1.274	13.422	22.2	4.27
120	1.692	13.004	21.0	3.9
130	2.221	12.475	20.0	3.6
140	2.887	11.809	19.46	3.25
150	3.716	10.980	18.50	2.9
160	4.739	9.957	18.00	2.6
170	5.990	8.706	17.60	2.2
180	7.510	7.186	17.50	1.8
190	9.336	5.360	17.40	1.3
200	11.525	3.171	17.15	0.78
210	14.123	0.573	17.05	0.15
212	14.696	0.000	17.00	0.00

(or concentration) in the contacting gaseous phase. This generalization is known as the Law of Henry and Dalton, and it will be so designated in future references.

The importance of the Law of Henry and Dalton to this study of steam for heating purposes is in the fact that it enables us to compute the amount of any gas which can be dissolved in water or condensate when we know the partial pressure of the gas in the contacting gaseous phase. The fact that only dissolved gases can have any influence on corrosion emphasizes the usefulness of the Law of Henry and Dalton. In the design of apparatus for the removal of dissolved gases from water the Law of Henry and Dalton has been used as the guiding principle and such apparatus endeavors to reduce the partial pressure of the dissolved gases in the gaseous phase to as low a value as is practicable.

The specific solubilities of oxygen and carbon dioxide differ. Water at 60 F

in contact with pure oxygen at 15 lb per square inch pressure is saturated when it has dissolved 34 milli-liters of oxygen per liter of water; in contact with pure carbon dioxide at the same pressure, water dissolves 1000 milli-liters of CO_2 per liter of water. In contact with air, under like conditions, $0.21 \times 34 = 7.1$ ml/liter of oxygen dissolves at saturation, and $0.003 \times 1000 = 0.3$ ml/liter of carbon dioxide.

The rate of solution to a point of saturation depends on the surface of water exposed to the gas and the distance the dissolved molecules of the gas must travel to produce homogenity of distribution. The greater the surface the greater the area exposed to impingement by the gas molecules and the shallower the vessel the shorter the distance each dissolved molcule must travel. Conversely the rate of degasification of water depends on exactly the same factors as well as the partial pressure of the gas in the contacting gaseous phase.

The specific solubility of a gas changes with temperature but the law of proportionality of solubility in the liquid to partial pressure of the gas holds true regardless of changing specific solubility values. At 212 F and a partial pressure of oxygen of 15 lb per square inch, the solubility of oxygen is 17 ml/liter; but if the partial pressure of oxygen is 1.5 lb the solubility at this temperature is only 1.7 ml/liter. Regardless of specific solubility values, the solubility of a gas is zero if its partial pressure in the gaseous phase in contact with the liquid is zero.

Carbon Dioxide

Carbon dioxide in steam or condensate in general follows the same rules that govern solubility of oxygen. The carbon dioxide content in any water may be present as dissolved carbon dioxide (CO_2), as bicarbonate Radical $(HCO_3)^-$ and as carbonate Radical $(CO_3)^=$ associated with a positive metallic radical. Since the bicarbonate Radical readily decomposes to form dissolved and finally gaseous carbon dioxide (CO_2), it is frequently referred to as carbon dioxide in the half-bound condition.

Bicarbonate Formation

It is pertinent at this point to discuss the mechanism of accumulation of available carbon dioxide in the raw water.

When carbon dioxide dissolves in water, some of it reacts with water thus:

$$\underset{CO_2}{\textit{Carbon Dioxide}} + \underset{H_2O}{\textit{Water}} \rightleftharpoons \underset{H_2CO_3}{\textit{Carbonic Acid}}$$

The carbon dioxide thus formed ionizes to some extent as follows:

$$\underset{H_2CO_3}{\textit{Carbonic Acid}} \rightleftharpoons \underset{H^+}{\textit{Hydrogen Ion}} + \underset{HCO_3^-}{\textit{Bicarbonate Ion}}$$

By so ionizing, it raises the hydrogen-ion concentration of the water. As the raw water passes along its course, it comes in contact with limestone ($CaCO_3$) and dissolves some of it. The limestone is relatively insoluble, but the small amount dissolved ionizes thus:

$$\underset{CaCO_3}{\textit{Limestone}} \rightleftharpoons \underset{Ca^{++}}{\textit{Calcium Ion}} \textit{ and } \underset{CO_3^{--}}{\textit{Carbonate Ion}}$$

With carbonic acid present, the next thing to occur is as follows:

Carbonate Ion and Hydrogen Ion → Bicarbonate Ion

$$CO_3^{--} \qquad H^+ \qquad HCO_3^-$$

Thus, since they unite to form an additional bicarbonate ion, both hydrogen ion and carbonate ion are removed from the water. When this has occurred, the water is no longer saturated either with limestone or carbonic acid, more of them dissolve, and so gradually the water accumulates a quantity of bicarbonate. Because all reversible chemical reactions will continue until a state of equilibrium is reached when the tendency for the reaction to proceed in either direction is equalized, the introduction of a secondary reaction which removes one or more of the products of the primary reaction upsets the balance of the original equilibrium. In the case of limestone, the secondary reaction and the primary reaction must both reach equilibrium before the dissolution of limestone

TABLE 2. DISSOLVED SOLIDS IN THE CROTON AND CATSKILL WATER AND OTHER CHARACTERISTIC WATERS

PARTS PER MILLION

	1	2	3	4
	Croton	Catskill	Relative Average Composition North American Surface Waters	City of Detroit
Bicarbonate (HCO_3)	36.6	15.9	67.89	96.5
Sulphate (SO_4)	8.2	3.2	15.31	16.4
Chloride (Cl)	5.0	1.3	7.44	7.0
Nitrate (NO_3)	0.6	0.6	1.15	....
Silica (SiO_2)	9.0	1.0	8.60	2.8
Iron (Fe)	....	....	0.45	0.6
Calcium (Ca)	9.6	5.2	19.36	26.2
Magnesium (Mg)	3.5	0.8	4.87	7.4
Sodium (Na)	3.5	1.2	8.50	4.4
Total Solids	57.4	21.1	99.0	112.3
Available Carbon Dioxide (CO_2)	26.4	11.5	49.0	69.5
Dissolved Unbound Carbon Dioxide at saturation[a] at 60 F	0.6	0.6	0.6	0.6
Dissolved Oxygen at saturation[a] at 60 F	10.0	10.0	10.0	10.0

[a] Saturated in contact with the atmosphere.

ceases. In such cases as the solution of iron, where one product of a secondary reaction is so insoluble that it can no longer have any practical effect, the original reaction continues until one or more of the ingredients is used up.

Thus solvent action along its course brings the water to the point of steam generation with a content of dissolved material the amount and character of which reflect the geographic locus of the drainage area whence it is derived. Table 2 illustrates this fact. Column 3 shows that in practically all North American surface waters the accumulation of bicarbonate goes on. From Columns 1 and 2 it is seen that the bicarbonate, or available carbon dioxide, in the Croton water is more than twice that in the Catskill; but by comparison

with Col. 4 it appears that Croton water has considerably less than half the bicarbonate in Detroit City water.

RESULTS OF SIMULTANEOUS SOLVENT ACTION OF WATER WITH ATMOSPHERIC GASES ON METALS

Carbon Dioxide

At *pH* values of 9.6 or greater, iron has practically no tendency to dissolve in water. At *pH* values less than this iron does dissolve and the tendency to dissolve increases as the *pH* value decreases.

Hence the dissolution of carbon dioxide in water accelerates the dissolution of iron therein, because the carbon dioxide on dissolving decreases the *pH* value of the water. In the same way the addition of any acid, as sulphuric acid, to the water, decreases the *pH* value therein and accelerates the dissolution of iron. The dissociation constants of acids vary with the acid and, in general, are an indication of the strength of the acid. Carbonic acid, formed by the solution of carbon dioxide in water, is a weak acid and dissociates but little in comparison with the strong acid, sulphuric. Carbonic acid, however, dissociates to produce a greater concentration of hydrogen ion than water and for this reason the *pH* value is lowered by the solution of carbon dioxide and the dissolution of iron is accelerated.

Although dissolved carbon dioxide has the same effect in accelerating the dissolution of iron, as has an acid like sulphuric, fortunately the quantity factor in the dissolution by dissolved carbon dioxide is much lower than that of sulphuric acid. Thus, when water containing sulphuric acid is in contact with iron, dissolution continues until the acid is entirely exhausted and the *pH* value has risen to 9.6. When dissolution occurs by reason of dissolved carbon dioxide, the *pH* value begins to rise immediately ferrous salt is present in solution, and the dissolving action is vastly retarded or stopped long before the quantity of iron dissolved is equivalent to the carbonic acid present.

Thus, though the dissolved carbon dioxide sets up in the water the low *pH* value that is accelerative to dissolution of the metal, there is no sustained dissolution thereof because the dissolving process itself supplies to the water the ferrous iron that increases the *pH* value and thus slows down the dissolving tendencies. It is axiomatic that at high concentration of carbon dioxide, more dissolution of iron must occur to effect this result, than at low concentration; and therefore corrollary that the concentration of carbon dioxide in the steam should be suitably restricted, so that any water condensed therefrom, and dissolving carbon dioxide therefrom in accordance with the Law of Henry and Dalton, should contain insufficient dissolved carbon dioxide to cause any substantial dissolution of the metal before a protective *pH* value is established in the water.

Because of the low quantity factor of corrosion by dissolved carbon dioxide limitation of the quantity in the steam to a value of 15 to 20 ppm. or thereabouts, is advisable, but advantages to be gained by any further diminution in carbon dioxide content are at a cost incommensurate with their value. The figure 15-20 ppm. is mentioned as an arbitrary standard, because it is obtained without great difficulty and because experience and theory agree that this amount of carbon dioxide in the absence of oxygen cannot be held responsible for corrosive effects worthy of consideration.

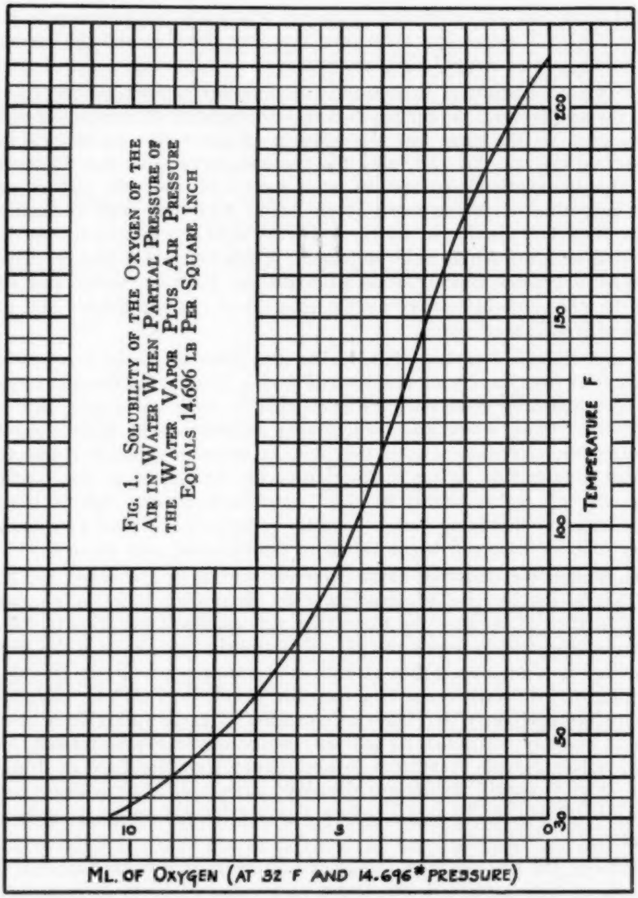


Fig. 1. Solubility of the Oxygen of the Air in Water When Partial Pressure of the Water Vapor Plus Air Pressure Equals 14.696 lb Per Square Inch

Oxygen

As iron dissolves in water, ferrous hydroxide and hydrogen are formed. When oxygen is simultaneously present, it unites with ferrous hydroxide to form ferric hydroxide, and with hydrogen to form water. The chemical reactions are as follows:

$$2\,Fe(OH)_2 + 1/2\,O_2 + H_2O \rightarrow 2\,Fe(OH)_3$$
$$H_2 + 1/2\,O_2 \rightarrow H_2O$$

Ferric hydroxide is red like ordinary iron rust and is almost insoluble. Thus dissolved oxygen, by uniting with the ferrous hydroxide, and changing it to insoluble ferric hydroxide, removes all dissolved iron from the solution and hence the water is as free to attack the metal as though it were freshly distilled pure water. Also, dissolved oxygen by uniting with the hydrogen removes it, and thus any presence of hydrogen does not interfere with the rapid further dissolution of iron. The fact must be emphasized that dissolved oxygen accelerates dissolution of iron by constantly removing the iron from solution, and for the same cause it sustains the dissolving of iron. Oxygen is used up in the process, and naturally, if there is only a limited supply of oxygen in the water it will finally be used up, whereupon ferrous hydroxide can accumulate, retard, and finally stop any dissolving action. But on the other hand, if the water is in contact with a gaseous source of oxygen, then as dissolution of the metal uses up the oxygen in the water, the gaseous supply of oxygen hastens to replenish the depleted supply in the water in accordance with the Law of Henry and Dalton.

Under such conditions there is no limitation to the amount of iron that can be corroded by any given amount of water. The process is continuous and the water merely acts as a contact agent to bring together the iron and the oxygen.

Carbon Dioxide and Oxygen

Carbon dioxide accelerates dissolution of iron by decreasing the pH value, and thus hastens replenishment of the ferrous iron removed from the water by union with oxygen. When the iron is thus steadily removed, the low pH value due to the carbon dioxide is steadily maintained, so that the rate of dissolution of iron remains high, and oxygen, by uniting with the iron and removing it from the water, is the sustaining factor of this high rate of dissolution. Hence, it follows that with both carbon dioxide and oxygen present, more rapid continuance of corrosion will occur than with either one singly; and again, if oxygen in gaseous form is in contact with the water to replenish, in accordance with the Law of Henry and Dalton, any oxygen used by uniting with iron, then the amount of corrosion that may occur at this high rate of dissolution is unlimited.

Pure water, or water containing dissolved carbon dioxide to a very limited extent, exercises a dissolving action on the metal, but in so slight a degree as to be negligible in the great majority of cases. However, when oxygen is present as a sustaining factor in this dissolution, then the limitation of dissolution that will occur depends entirely upon the total oxygen available to combine with the dissolved iron and thus sustain the reaction.

ANALYSIS OF CONDITIONS IN THE STEAM GENERATING, DISTRIBUTING AND UTILIZING SYSTEMS

Problems of the Steam-Generating System

The steam-generating system must (1) develop steam sufficiently free from association with oxygen and carbon dioxide to meet the requirements of the remainder of the system; (2) establish and maintain conditions in the water such that its contacts with the surfaces of the generating system will not result in interruption of its continuity of operation, or effect deterioration; and (3) prevent entrainment by the steam of boiler water or solids, which by deposition in valves or lines in the distribution or utilizing systems, may cause trouble.

Separation of Associated Gases—Oxygen

If air-saturated water is heated in a vessel with restricted outlet, the oxygen content will decrease because the specific solubility decreases with rise in temperature and because the vapor pressure of water increases with temperature. As the vapor pressure of water increases the partial pressure of water vapor increases, and, therefore, the partial pressure of oxygen in the gaseous phase must decrease thus automatically decreasing the solubility of oxygen. In Fig. 1 the solubility of oxygen at increasing temperature and decreasing partial pressure, because of the increasing partial pressure of water vapor, is shown for the full range of water temperatures under atmospheric pressure. At 212 F the vapor pressure of water is equal to the atmospheric pressure so that the partial pressure of oxygen must be zero and, therefore the solubility of oxygen is zero.

Table 3 gives values of actual tests on feed water in an open, vented, feed-water heater. These values are sometimes larger, sometimes smaller than the values shown in Fig. 1 for the corresponding temperature. The rate at which water and low pressure steam are supplied to such heaters varies with the demand on the heater and with the supply of exhaust steam respectively. Obviously the time consumed by the water in passing through the heater varies and this variation is apparently sufficient to shorten, during some periods, the time below the minimum required by the molecules of dissolved oxygen to travel to the nearest liquid surface and escape into the vapor phase.

If the heater is provided with sufficient excess steam to maintain continuously the water temperature at 210 to 212 F, and to vent from the proper point in the heater, a quantity of steam sufficient to insure a partial pressure of oxygen as nearly zero as may be, then the oxygen content of the effluent water will not exceed 0.3 mil./liter and perhaps less. While 0.3 ml./liter is not as low an oxygen content as is guaranteed with some types of deaerating heaters (0.025 ml./liter) it is the authors' belief (because of the Law of Henry and Dalton) that this degree of removal is ample for the purposes of steam heating. The records of the generating system showing maintained heater-temperature, and of uninterrupted ample venting, constitute unimpeachable evidence as to the oxygen content of the steam it is furnishing.

Solution of the problem of oxygen-removal whether at the heater in the generating system, or in the utilizing system where inleakage of air is occurring, rests upon the fundamental factors of providing the very minimum of partial pressure of oxygen in gases making contact with the water, and providing a very short path for the dissolved oxygen molecules to the contacting

surfaces. Practice must recognize the second, or time factor, although theory holds the first factor sufficient in itself for the removal of oxygen. The partial pressure factor at the heater is controlled by adequate supply of steam and venting; the time factor is controlled by spraying the water, or by passing it over trays in thin layers, to provide a short path of escape for the dissolved oxygen molecules. In the utilizing division there is only crude control of the partial-pressure factor and no control of the time factor. A pump may intermittently control the partial pressure of oxygen in the utilizing system but the rate at which oxygen is absorbed by the condensate when the steam is condensed far exceeds the rate at which oxygen is liberated by the main stream of condensate even though the partial pressure of oxygen in the contacting vapor phase is reduced.

TABLE 3. TEMPERATURE AND OXYGEN CONTENT OF RAW AND FEED WATER—KIPS BAY STATION

Date	Raw Water		Feed Water	
	Temp. (Deg. Fahr)	Oxygen ml./liter	Temp. (Deg. Fahr)	Oxygen ml./liter
12/11	48	8.0	170	0.7
2/5	39	8.0	172	1.7
12/29	46	8.1	176	0.6
12/2	48	8.0	180	0.6
4/8	40	8.0	180	1.0
1/28	40	6.4	180	1.06
2/11	39	7.0	180	1.3
2/18	38	7.9	181	1.6
6/26	68	8.0	182	1.0
11/7	62	7.8	186	0.5
11/21	58	8.0	186	0.5
6/13	60	7.9	186	0.9
1/22	40	6.3	186	0.95
8/27	76	6.1	186	1.0
3/4	44	7.8	186	1.06
11/14	58	8.0	188	0.6
3/25	42	6.0	188	0.7
6/4	60	7.9	188	1.0
9/18	74	7.4	190	0.5
4/22	48	7.9	190	1.2
4/1	42	8.1	190	1.3
3/11	40	7.7	192	1.0
1/7	44	7.6	195	0.36
8/13	72	7.8	195	0.5
9/9	74	7.5	196	0.4
7/3	65	7.9	198	1.0
9/4	72	7.6	200	0.5
2/25	40	7.8	200	1.2
9/10	74	7.8	202	0.6
4/29	50	8.1	206	1.0
6/19	68	7.9	208	0.9
5/6	52	8.0	210	0.7
5/13	54	7.8	210	0.7
4/15	46	7.1	210	1.3

Separation of Associated Gases—Carbon Dioxide

In the district steam plant and in the industrial plant with a large per cent of make-up, it is usually uneconomical to evaporate and effectively degasify all

raw-water make-up in order to eliminate completely carbon dioxide. The slightly more complete exclusion of carbon dioxide by this process than by judiciously chosen treatment offers no particular advantages commensurate with the cost. It should be noted that the percentage of raw discriminately treated New York City make-up water has too little effect on the carbon dioxide content of the steam made therefrom to place the steam at any disadvantage for heating purposes as compared with steam made from water containing an equivalent amount of heating returns.

In waters which contain as much or more bicarbonate than that shown by Detroit City water the available carbon dioxide content can usually be considerably reduced by pretreatment with lime and soda. In such cases, the authors believe that lime-soda pretreatment represents the most economical solution to decreasing the available carbon dioxide, as well as to best prepare the water for boiler feed water. There is, however, only a slight gain to be realized by pretreatment of Croton water with lime and soda because the available carbon dioxide is reduced by reactions at the heater without pretreatment almost as much as it would be by pretreatment. The available carbon dioxide in the Catskill water is too low naturally to be improved by treatment.

TABLE 4. DISSOLVED SOLIDS IN THE CROTON AND CATSKILL WATER AND OTHER CHARACTERISTIC WATERS AFTER LIME-SODA OR ZEOLITE TREATMENT

PARTS PER MILLION

	Croton		Catskill		Rel. Average Comp. N.A. Surface Waters		City of Detroit	
	Lime-Soda	Zeolite	Lime-Soda	Zeolite	Lime-Soda	Zeolite	Lime-Soda	Zeolite
Hydroxide (OH)	1.7	0.0	1.7	0.0	1.7	0.0	1.7	0.0
Carbonate (CO_3)	18.0	0.0	18.0	0.0	18.0	0.0	18.0	0.0
Bicarbonate (HCO_3)	0.0	36.6	0.0	15.9	0.0	67.9	0.0	96.5
Sulphate (SO_4)	8.2	8.2	3.2	3.2	15.3	15.3	16.4	16.4
Chloride (Cl)	5.0	5.0	1.3	1.3	7.4	7.4	7.0	7.0
Nitrate (NO_3)	0.6	0.6	0.6	0.6	1.2	1.2	....	....
Silica (SiO_2)	9.0	9.0	1.0	1.0	8.6	8.6	2.8	2.8
Iron (Fe)	....	....	....	....	0.5	0.5	0.6	0.6
Calcium (Ca)	6.0	2.0	6.0	2.0	6.0	2.0	6.0	2.0
Magnesium (Mg)	0.8	0.5	0.8	0.5	0.8	0.5	0.8	0.5
Sodium (Na)	15.0	17.9	10.3	5.5	20.2	35.0	20.0	45.7
Total Solids	64.3	61.2	42.9	21.9	79.7	103.9	73.3	122.5
Available Carbon Dioxide (CO_2)	13.2	26.4	13.2	11.5	13.2	49.0	13.2	69.5

No decrease of available carbon dioxide occurs with pretreatment by zeolite alone. The zeolite mineral effectively removes the calcium and magnesium from the water but does not remove the bicarbonate. Excessive quantities of bicarbonate in the feed water result in, among other things, excessive carbon dioxide in the steam and therefore subsidiary lime treatment and acid treatment, or the latter alone, are required together with all the accompanying niceties

of control that distinguish acid treatment wherever used, when a bicarbonate water is treated with zeolite.

Table 4 shows the results of treating the waters of Table 2 with lime-soda and with zeolite. The effect of the pretreatment on the bicarbonate and therefore the relative effect on the carbon dioxide content of the steam is clearly evident.

As the temperature of a bicarbonate-containing water is raised, the tendency of the bicarbonate is to revert to the carbonate, and at the temperature of normal boiling reversion proceeds with considerable speed. If sufficient time were given under the conditions prevailing at the heater, reversion would be complete, and the only remaining available carbon dioxide in the water would

TABLE 5. DISSOLVED GASES IN STEAM DERIVED FROM CROTON WATER

South Main

Date	Oxygen ml./liter	Carbon Dioxide Parts per Million	*pH* Value of Condensate
April, 1930			
4	0.50	20.6	4.9
5	.45	21.9	4.7
8	.41	19.7	5.2
9	.38	18.6	4.9
10	.34	18.1	4.8
11	.46	19.8	5.0
14	.45	20.8	4.9
15	.34	18.1	5.2
16	.45	19.7	5.9
17	.41	19.7	6.0
	North Main		
August, 1930			
5	0.34	16.3	5.0
6	.32	15.6	5.0
7	.39	21.3	5.2
8	.35	16.3	5.2
12	.49	18.5	5.0
13	.42	17.3	5.0
15	.28	20.3	5.0
26	.35	17.5	5.0

be in the form of normal carbonate from which loss of carbon dioxide occurs considerably less rapidly than from the bicarbonate.

At the heater, therefore, and particularly under the conditions of maintained temperature and ample venting of steam, much of the bicarbonate will revert to carbonate with very considerable reduction of the available carbon dioxide passing on to the boilers. Jackson and McDermet (*J. Ind. Eng. Chem.* Vol. 15, pp. 959-61, 1923) have determined that by deaeration carbon dioxide loosely combined as bicarbonate is removed to the extent of about thirty per cent. That this is the case with Croton water is shown by the figures of Table 5, in which the maximum carbon dioxide found in the steam was 21.9 ppm. and minimum 15.6, while the general figure for available carbon dioxide in the feed water was 26.4 ppm. (Table 3, Column 1). Most of this decrease in available

carbon dioxide is due to its separation at the heater; however, some small amounts separate in the form of economizer deposits and boiler sludge as illustrated by Table 6. Then again, all available carbon dioxide in the boiler water does not pass into the steam, as there is always some carbonate in the boiler water as illustrated by Table 7, which lists the average concentrations for the month of December, 1930, in a phosphate-treated boiler water from a large boiler operating at high ratings on Croton River water.

Discussion of water conditioning for the specific purpose of protecting the boiler surfaces is beyond the scope of this report, and reference is made to papers directed thereon: Mumford, *Combustion,* Vol. 1, pp. 43-46, Sept. 1929;

TABLE 6. ELIMINATION OF CARBON DIOXIDE IN DEPOSITS AND SLUDGES—ANALYSES IN PER CENT

	Sludge from Water Drum	Scale Water Drum	No. 3 Economizer Deposit
Carbon Dioxide (CO_2)	4.75	14.41	14.83
Sulphur Trioxide (SO_3)	3.14	10.70	1.36
Phosphorous Pentoxide (P_2O_5)	24.04	5.26	17.73
Silica (SiO_2)	9.74	6.70	6.54
Ferric Oxide (Fe_2O_3)	3.10	2.84	1.24
Aluminum Oxide (Al_2O_3)	....	0.33	....
Calcium Oxide (CaO)	38.06	39.50	45.52
Magnesium Oxide (MgO)	10.38	15.75	7.70
Sodium Oxide (Na_2O)	....	2.80	....
Loss 105 C	0.45	0.65	0.30
Net Ignition Loss	6.15	0.14	4.80

TABLE 7. AVERAGE COMPOSITION OF BOILER WATER—PARTS PER MILLION
No. 5 BOILER

December 1930

Sodium Hydroxide ($NaOH$)	260
Sodium Carbonate (Na_2CO_3)	143
Sodium Phosphate (Na_3PO_4)	252
Sodium Sulphate (Na_2SO_4)	1450
Sodium Chloride ($NaCl$)	Undetermined

Markson, *Combustion,* Vol. 1, pp. 27-30, April 1930; Hall, *Iron and Steel Engr.,* Vol. 6, pp. 380-389 (1929); *Mech. Engr.,* Vol. 48, 317-327 (1926); *Trans. A. S. M. E.,* Fuels and Steam Power Sections, Vol. 50, No. 33, pp. 65-75 (1928).

Prevention of Entrainment of Boiler Water in the Steam

If boiler water is entrained in steam a quantity of dissolved or suspended matter is carried with the water which depends on the amount of water entrained and the concentration of solids in the boiler water. The distance from the steam generator to which such solids may be carried depends on the steam velocity and on the frequency of impingements which have a coalescing and separating effect and is of interest because of its effects in the distributing and utilizing divisions, particularly the latter, because deposits therein primarily due to other causes may be mistakenly ascribed to such carry-over.

In Table 8 is presented an analysis (No. 1) of sludge characteristic of that

occurring in phosphate-treated boiler water; and analyses (Nos. 2-7) of various deposits taken from the mains of the distributing system. That all are largely or in part derived from carry-over of boiler water in the steam is obvious because of the phosphate content for which no other source is possible. Some of them, notably Nos. 2, 4 and 7 in particular, show admixture of other material with the boiler sludge, as shown by the very considerable amount of silica they contain; and others, notably Nos. 4, 5, 6 and 7, by their magnetic quality, and their content of iron oxide, show accretion of corrosion products from some source—probably the trap, in view of the conclusions developed in the discussion of Division 2.

In Table 9 are presented analyses of several samples of condensate from various points in the system. Those from the street traps contained considerable suspended and dissolved solids; those taken from Building *A* contained but little dissolved and suspended solids, indicating that any carry-over of boiler water in the steam is largely removed in the traps on the distributing mains before the steam enters the building. In practically all cases, phosphate was present; and likewise iron. Further on, in the analyses of Table 12, will be found the relative quantities of iron oxide and phosphate in samples of deposits taken from various points in Building *A*. The high content of iron oxide and low content of phosphate in these deposits is indubitable proof that their major source is not boiler-water sludge but the dissolution of the piping system of the building itself, and its transformation to the red oxide by the inseepage of the atmospheric gases into that system.

While everything, therefore, points to the fact that in this case carry-over in the steam has little or no relation to the problem of deposits beyond the street meter, the prevention of carry-over has been and is vitally necessary to the generating system for its own economic operation, and totally independent of its relations to the utilizing system. The reasons need not be enumerated here, but are inherent in conditions in the generating station, and in the distributing mains.

As the scope of this report does not permit general discussion of foaming and priming, reference is made to publications thereon. (Hall, *N. E. L. A.* Pub. No. 278-81, *Journal American Water Works Association,* Vol. 21, pp. 79-100; *Trans. A. S. M. E.* Fuels and Steam Power Sections, Vol. 50, No. 33, pp. 65-75.) Discussion of the influence on the Kips Bay boilers of boiler-water alkalinities, cleanliness, or concentrations in their relation to carry-over; or the mechanical features of boiler baffling, installation of separators and continuous blow-down, control of water levels, drainage of steam drum, is a matter of record in the publications of Mumford (*Combustion,* September 1929, and *Trans. A. S. M. E.* Fuels and Steam Power, Vol. 51, No. 22, pp. 363-374) and Markson (*Combustion,* April 1930).

The problem of carry-over is receiving attention to a degree increasing with the capacity of the steam generating units. The knowledge on which the solutions of the problem of carry-over could be based must be built up by the engineers responsible for the operation of these new generators because the changes in the generators have so changed the nature of the problem that knowledge based on earlier installations is not usually sufficiently profound to be applicable. Breadth of general experience, however, is giving brevity to the time required for meeting any specific set of conditions.

TABLE 8. BOILER SLUDGE AND DEPOSITS TAKEN FROM TRAPS ON DISTRIBUTION MAINS—ANALYSES IN PER CENT

	1	2	3	4	5	6	7
	Boiler Sludge No. 5 Boiler	Gate Valve Deposit 1st Ave. Main Distributing Line	North Trap West Set, 622 1st Ave.	South Trap 37th St. E. of 3rd Ave.	South Trap 41st St. E. of Park Ave.	Service Trap on 150 lb Header and *C* Meter	Six Inch Valve 90 lb Header
Sulphur Trioxide (SO_3)	1.3	Trace	0.6	0.3	0.6	0.8	1.9
Carbon Dioxide (CO_2)	1.6			included in Ignition Loss			
Phosphorous Pentoxide (P_2O_5)	31.5	29.9	33.3	7.6	27.1	19.2	9.8
Silica (SiO_2)	4.6	12.1	7.6	32.9	8.6	11.5	16.9
Ferric Oxide (Fe_2O_3) } Aluminum Oxide (Al_2O_3) }	0.6	1.2					
Ferric Oxide (Fe_2O_3)			8.1	39.3	27.3	41.0	17.6
Aluminum Oxide (Al_2O_3)			1.0	4.4	1.8	2.3	4.0
Calcium Oxide (CaO)	36.3	37.8	35.7	7.9	26.2	20.7	17.4
Magnesium Oxide (MgO)	17.7	13.6	13.5	2.7	5.2	3.1	23.1
Loss at 105 C	2.7	0.5	1.4	0.8	1.2	1.0	2.8
Ignition Loss	4.3	5.7	4.1	7.2	2.0	1.6	12.8
	non-magnetic	non-magnetic	non-magnetic	magnetic	slightly magnetic	slightly magnetic	magnetic

TABLE 8A. DESCRIPTION OF SAMPLES OF TABLE 8

1. *Boiler Sludge*
   Tan powder, easily pulverized. Non-magnetic.
2. *Gate Valve Deposit, 1st Ave. Main Distributing Line*
   White deposit readily pulverized to impalpable powder. Non-magnetic.
3. *North Trap, West Set. 622 1st Ave.*
   Very soft, light gray powder, non-magnetic. Amount of deposit 40 grams.
4. *South Trap, 37th St. East of 3rd Ave.*
   Dark gray, soft powder, magnetic. Amount 30 gms.
5. *South Trap, 41st St. East of Park Ave.*
   Light brown powder, slightly magnetic. Amount, 10 gms.
6. *Service Trap on 150-lb Header and C Meter.*
   Soft mud on sides and bottom of trap. When dry, light brown, slightly magnetic powder. Amount not recorded.
7. *Six-Inch Valve, 90-lb Header*
   Light tan, scaly deposit on downstream side of gate. Very small amount, magnetic light gray, soft. Amount, about 1.5 gms.

Table 9. Analyses of Condensate Taken from Street Manholes and Traps from Building A—Parts Per Million

Location	Date of Sampling	Suspended Matter	Dissolved Solids	Phosphate PO_4	Iron
Street trap, 622 1st Ave.	9/8/30	137[a]	456[b]	present	present
Street trap, 37th St. E. of 3rd Ave.	9/8/30	419	30	present	present
Street trap, 41st St. E. of Park Ave.	9/8/30	9717	56	present	present
Trap at Meter	Shortly prior to April '30	88	36	25	not recorded
Meter in Building	4/1/30	Total Solids 3 ppm.	..	none found	present
Sample by cracking valve on trap at main 90-lb. header	4/1/30	Total Solids 1.6 ppm.	..	trace	none found
Body of *C* Meter in Bldg	8/5/30	1	7	none found	present
Trap of *C* Meter in Bldg	8/8/30	1	5	none found	present
Trap of main 90-lb. header	8/7/30	4	12	present	present
8-in. H.P. riser from 90-lb. header	8/6/30	0	3	none found	present
Inlet side trap at heel of 8-in. riser	8/25/30	1	3	present	present
Return hot water heater on 29th floor. Before trap	8/12/30	1	9	present	present
Return hot water heater on 29th floor. After trap	8/13/30	35	14	present	present
Kitchen Uptown Club, 26th floor. Returns from Coffee Urns	8/15/30	1	4	present	present
19th floor. Intermediate System	4/1/30	Total Solids 2.6 ppm.	..	none found	none found
Low pressure returns before vacuum pump	8/25/30	0	18	none found	present

[a] Result is low because of colloidal nature of some of the suspended matter making it impossible to obtain perfectly clear filtrate.
[b] Result is high because of colloidal nature of some of the suspended matter carried into filtrate.

By maintaining maximum feed-water temperature as the water is heated in a well vented feed-water heater the generating system assures steam sufficiently free from association with oxygen and carbon dioxide to meet the requirements of the remainder of the system. By careful control and unceasing attention to the maintenance of proper chemical relations in the boiler water, the generating system assures uninterrupted supply of steam and minimum deterioration of piping and heating surfaces. By development work the generating system is approaching the solution of the problem of carry-over introduced by the pressure of economic development of steam generating units.

PROBLEMS OF THE DISTRIBUTING SYSTEM

By definition the distributing system is that part of the system where, uninterrupted by seasonal or operative conditions, any leakage is from the system into the surrounding atmosphere except only those infrequent times when shut downs are made for repairs or examinations.

Because any corrosive action occurs only after the steam condenses into water, it is necessary to understand thoroughly the character of the steam and the distribution of its components in any condensate that forms. The determination of the different components is accomplished with the use of the best laboratory methods available and in the case of oxygen is exact but in the cases of *pH* value and carbon dioxide the available methods do not permit the same order of accuracy as with oxygen. Because of the low capacity factor of pure water for dissolving iron we can consider oxygen as the capacity factor in corrosive action and the *pH* value as the velocity factor.

The Velocity Factor—pH Value and Carbon Dioxide

The *pH* value of entirely pure water is approximately 7 when measured at room temperature. The value actually obtained in careful distillation of water, as for instance in the laboratory still, ranges from about 5.5 to 6.5. Only by the most careful manipulation of such water following its condensation to provide certain removal of all gases such as carbon dioxide and sulphur dioxide, which are acid anhydrides, will the value of 7 be obtained (see Acree and Fawcett, Loc. cit.). Thus the *pH* values of Table 11, Section 4, taken in 1928, show that the condensate had acquired alkalinity from some source, and of course, most probably by carry-over of alkaline boiler water in the steam. The lower *pH* values of the other data of more recent months are therefore proof of lesser, if any, appreciable quantity of alkaline carry-over in the steam reaching the utilization system.

By definition, a *pH* value of 7 represents neutrality, above 7, alkalinity, and below acidity. As noted, distilled water usually lies on the acid side, as indicated by the *pH* value of 5.5 to 6.5, although it should be remembered that in an unbuffered water of this sort, the correct determination of *pH* value represents no simple task, and an error of several tenths or even a unit may readily be introduced into the determination.

The samples of Table 5, and those of Table 11 designated *steam* samples were obtained by condensing the steam in a cooling coil, particular attention being paid that all gases in the steam should dissolve as condensation occurred. The *pH* values of those samples range from a minimum of 4.7 to a maximum of 6, with a general average of 5.1. Excepting from Table 11, No. 2 of

TABLE 10. SOLUBILITY OF OXYGEN AND CARBON DIOXIDE AT DIFFERENT PARTIAL PRESSURES IN THE CONDENSATE FROM STEAM AT THE ABSOLUTE PRESSURES OF 165 LB AND 14.696 LB[a]

Oxygen

1. Steam pressure (lb absolute)	165	165	14.696	14.696
2. Temperature (F)	366	366	212	212
3. Vol. 1 lb water as saturated steam (cu ft)	2.748	2.748	26.82	26.82
4. Oxygen in sample condensed with cooling coil (ml/liter)	0.3	1.0	0.3	1.0
5. Solubility assumed at 15 lb partial pressure (ml/liter)	16	16	17	17
6. Partial pressure of oxygen (lb)	0.000043	0.000144	0.0000034	0.0000115
7. Solubility in the condensate (ml/liter)	0.000046	0.000153	0.0000039	0.000014

[a] Since these calculations are based on approximate data, the value 15 lb has been used as atmospheric pressure at the higher pressure used as an example.

Carbon Dioxide

1. Steam pressure (lb absolute)	165	14.696
2. Temperature (F)	366	212
3. Vol. 1 lb water as saturated steam (cu ft)	2.748	26.82
4. Carbon Dioxide in sample condensed with cooling coil (ppm)	19.6	19.6
5. Solubility assumed at 15 lb partial pressure (ppm)	390	490
6. Partial pressure of Carbon Dioxide (lb)	0.00147	0.00012
7. Solubility in the condensate (ppm)	0.038	0.004

Section 1, regarding which some doubt exists; and Nos. 9 of Section 1 and Nos. 3 and 7 (second steam sample) of Section 2, in which the accumulation of carbon dioxide following partial condensation of the steam is plainly in evidence, the carbon-dioxide content of these samples range from a minimum of 13 ppm. to a maximum of 28.3 ppm. with a general average of 19.6 ppm.

The samples of Table 11 designated as *condensate,* were also passed through the cooling coil, but condensation had occurred in contact with steam as vapor phase and later contact had been made with the vapor phase present in the traps. The *pH* values of these samples (exclusive of Section 4 of Table 11) vary from 4.9 to 6.7, with general average of 5.8; the carbon-dioxide content varies from 0 to 22.3 ppm. with a general average of 7.6 ppm.

As the evaluation of these very small quantities of carbon dioxide is based upon the color change of an indicator, too great emphasis must not be placed on the numerical values. However, the content of carbon dioxide is markedly lower in the *condensate* than in the *steam* samples, as it should be to be in accord with the Law of Henry and Dalton. The higher *pH* values of the *condensate* samples show that the actual condensate which contacts the surfaces is of more desirable characteristics in this regard than would be considered the case if conclusions were based on the *steam* samples, as is frequently done.

The boiler water at Kips Bay is always maintained alkaline, and hence there is no possibility that acid anhydride other than carbon dioxide will be associated with its steam. This is the condition that obtains very generally in all steam derived from carefully conditioned boiler water. In the consideration, therefore, of any corrosive effects arising from *pH* value below neutrality (for instance, 5.1 for condensed steam and 5.8 for condensate), and unassisted by any quantity factor, the discussion may be limited to the development due to carbon dioxide.

Conditions at the Internal Surfaces

Any corrosive action by the steam occurs only after its condensation into water. Since the rate factor of dissolution is dependent on the *pH* value of such condensate and since any *pH* value decrease below 7 is dependent in turn on amount of dissolved carbon dioxide, it is necessary to find how much carbon dioxide can dissolve in condensate contacting the steam under consideration which contains 19.6 ppm. of carbon dioxide.

TABLE 10A. CALCULATIONS FOR TABLE 10

Oxygen

—1—

(*a*) Steam Pressure—150 lb gage
Oxygen Content—corresponding to 0.3 ml/l. in a condensed sample drawn through cooling coil, thus retaining all oxygen. At 150 lb gage pressure, 1 lb water = 2.748 cu ft saturated steam.

O_2 = 0.3 ml/l. = 0.42 ppm. = .00000042 lb per pound water.
At 0 deg C. and 760 mm 1 lb O_2 = 317000 ml. = 11.2 cu ft.

For 0.3 ml/l. = .00000042 lb O_2 per pound water.
$0.00000042 \times 1.71 = 0.00000072$ cu ft.

Partial pressure of O_2 at 150 lb gage pressure

$$\frac{0.00000072}{2.748} \times 165 = 0.000043$$

At 150 lb gage pressure = 366 F = 185.6 C.

$1 \text{ lb } O_2 = 11.2 \times \frac{273 + 185.6}{273} \times \frac{15}{165} = 1.71$ cu ft.

Solubility of O_2 in water at this partial pressure—

Assume solubility = 16 ml/l. at 185.6 C and 14.696 lb O_2.

$\frac{16 \times 0.000043}{15} = 0.000046$ ml/l. = 0.000066 ppm.

(*b*) $O_2 = 1.0$ ml/l. = 1.4 ppm. = 0.0000014 lb O_2 per pound water.

$0.0000014 \times 1.71 = 0.0000024$ cu ft.

Partial pressure:

$\frac{0.0000024}{2.748} \times 165 = 0.000144$ lb.

$16 \times \frac{0.000144}{15} = 0.000153$ ml/l. = .00022 ppm.

—2—

(*a*) At 14.696 absolute pressure = 0 gage pressure

1 lb water = 26.82 cu ft saturated vapor

At 212 F = 100 deg C and 14.696 lb pressure

$1 \text{ lb } O_2 = 11.2 \times \frac{373}{273} \times \frac{14.696}{14.696} = 15.3$ cu ft.

For 0.3 ml/l. = 0.00000042 lb O_2 per pound water

$0.00000042 \times 15.3 = 0.0000064$ cu ft.

Partial pressure at 14.696 lb.

$\frac{0.0000064}{26.82} \times 14.696 = 0.0000034$ lb.

Sol. $O_2 = 17$ ml/l. at 212 F and 14.696 lb O_2

$17 \times \frac{.0000034}{14.696} = 0.0000039$ ml/l. = .000006 ppm.

(*b*) For ml/l. of O_2, solubility is 0.000014 ml/l. = .00002 ppm.

Carbon Dioxide

—1—

Steam Pressure—150 lb gage
Carbon Dioxide—19.6 ppm.
At 150 lb gage pressure, 1 lb water = 2.748 cu ft saturated steam
$CO_2 = 19.6$ ppm. = .0000196 lb per pound water
At 0 deg C and 760 mm. 1 lb $CO_2 = 231000$ ml. = 8.2 cu ft.
At 150 lb gage pressure = 366 F = 185.6 deg C.

$1 \text{ lb } CO_2 = 8.2 \times \frac{273 + 185.6}{273} \times \frac{15}{165} = 1.25$ cu ft.

For 19.6 ppm. = .0000196 lb CO_2 per pound water
$0.0000196 \times 125 = .0000245$ cu ft.

Partial pressure of CO_2 at 150 lb gage pressure

$\frac{0.0000245}{2.748} \times 165 = 0.00147$ lb.

Solubility of CO_2 in water at this partial pressure—

Assume sol. $CO_2 = 390$ ppm. at 185.6 deg C and 14.696 lb CO_2

$390 \times \frac{0.00147}{15} = 0.038$ ppm.

—2—

At 14.696 lb abs. pressure = 0 gage pressure
1 lb water = 26.82 cu ft sat. vapor.
At 212 F = 100 deg C and 14.696 lb pressure

$$1 \text{ lb } CO_2 = 8.2 \times \frac{373}{273} \times \frac{14.696}{14.696} = 11.2 \text{ cu ft}$$

For 19.6 ppm. CO_2 = 0.0000196 lb per pound water
0.0000196 × 11.2 = 0.00022 cu ft

Partial pressure at 14.496 lb

$$\frac{0.00022}{26.82} \times 14.696 = 0.00012 \text{ lb}$$

Sol. CO_2 = 490 ppm. at 212 F and 14.696 lb CO_2

$$490 \times \frac{0.00012}{14.696} = 0.004 \text{ ppm.}$$

Carbon dioxide obeys the Law of Henry and Dalton under the conditions obtaining in the steam lines. Since the solubility of carbon dioxide is known (approximately) for the partial pressure of 14.696 lb (or one atmosphere) over a considerable temperature range, the amount of carbon dioxide in solution in condensate formed from and in contact with steam at any chosen pressure can be calculated, if the concentration of carbon dioxide in the steam is known. For example, assume that the laboratory test on a sample obtained by condensing steam of 165 lb absolute or 150 lb operating pressure in a cooling spiral gives the result of 19.6 ppm. of carbon dioxide. With each pound of steam, therefore, there is associated 0.0000196 lb of carbon dioxide. The volume of 1 lb of carbon dioxide at 366 F which corresponds to the operating pressure of 165 lb absolute is 1.25 cu ft. The volume of 0.0000196 lb of carbon dioxide for these conditions is 0.0000196 × 1.25 = 0.0000245 cu ft. The volume of 1 lb of water for the same conditions is 2.748 cu ft of saturated steam. Thus, 0.0000245 cu ft of carbon dioxide is associated with 2.748 cu ft of steam at absolute pressure of 165 lb, hence the partial pressure of carbon dioxide is

$$\frac{0.0000245}{2.748} \times 165 = 0.00147 \text{ lb.}$$

The solubility of carbon dioxide at the temperature of 366 F and a partial pressure of 14.696 lb is approximately 390 ppm. Hence at partial pressure of 0.00147 lb the solubility of carbon dioxide is

$$\frac{0.00147}{15} \times 390 = 0.038 \text{ ppm.}$$

These calculations and data are summarized in Table 10A. Calculations are also given for steam at atmospheric pressure. Thus with 19.6 ppm. of carbon dioxide present in the steam, any condensate in the 150-lb lines is saturated by 0.038 ppm. of dissolved carbon dioxide. In the lines with steam at atmospheric pressure, saturation of condensate occurs at 0.004 ppm. At intermediate pressures, the saturation values are intermediate of these values. The question of points at which the steam is stagnant and condensing, is considered in the discussion of oxygen.

The conclusion is obvious and is supported and emphasized by the data on

condensate samples in Table 11. So long as the partial pressure of the carbon dioxide remains of the order of magnitude noted in line 7, Table 10, enough dissolution of the gas will not occur in the condensed steam to do any damage whatsoever. Doubling or trebling of the carbon dioxide content used as example, still would leave the partial pressure in this rang. This relationship holds throughout the entire region of surfaces exposed uninterruptedly to the steam of pressure equal to atmospheric or greater.

Thus any deductions based directly on the *pH* values and carbon dioxide content of samples condensed with special attention to retention of the gases are inapplicable to the internal surfaces as defined in the preceding paragraph. They are applicable only at points of condensation, where said condensation occurs under circumstances akin to those surrounding the taking of samples.

pH Value in Absence of Oxygen

As noted heretofore in the section on results of simultaneous action of water on iron and carbon dioxide, in the absence of the quantity factor, oxygen, the tendency of the metal to dissolution is increased by lowered *pH* value, but so long as this lowering is effected by the dissolved carbon dioxide, the capacity of the water for dissolution of the metal is small because the dissolved iron itself readily builds the *pH* value to protective proportions. It is this fact in conjunction with the small partial pressure of the carbon dioxide, and therefore its slight solubility in the condensate, which has allowed the degree of carbon-dioxide removal to be a somewhat secondary consideration. The right thing to do, of course, is to restrict the carbon dioxide as much as economically possible, in order to prevent, so far as possible, any small dissolution, and also to avoid any danger of augmented rate and quantity of dissolution in case any concentration of oxygen is present for any reason.

THE QUANTITY FACTOR—DISSOLVED OXYGEN

The possibilities in deoxygenation of feed water consist of (1) use of deaerating heater, which if operating correctly, mechanically reduces the dissolved oxygen content to 0.025 to 0.05 ml. per liter; (2) use of open, direct contact heater, with temperature maintained at 212 F (if necessary, by means of live steam bled thereto) and with sufficient steam passing the vent to maintain low partial pressure of oxygen in the heater, in which case the dissolved oxygen content is reduced to 0.1 to 0.3 ml. per liter; (3) use of open, direct-contact heater, with no special precautions regarding maintenance of temperature or venting, in which case the dissolved-oxygen content of the effluent water may vary from 0.1 ml. upward, sometimes reaching 2 or 3 ml. per liter; (4) use of closed heater, in which case all oxygen dissolved in the feed water passes to the boiler and thence to the steam. In (1) and (2), a further step in removal of oxygen may be made by chemical fixation thereof following the heater, reducing it to zero if desired.

What degree of deoxygenation is essential for the production of steam that is entirely satisfactory for heating purposes? That question must be answered by the possibilities of corrosion that will ensue when the steam condenses, since corrosion of the system is not caused by the steam, but occurs only as the condensate simultaneously in contact with metal and gases, especially oxygen, makes possible their reaction with each other in dissolved form. Since

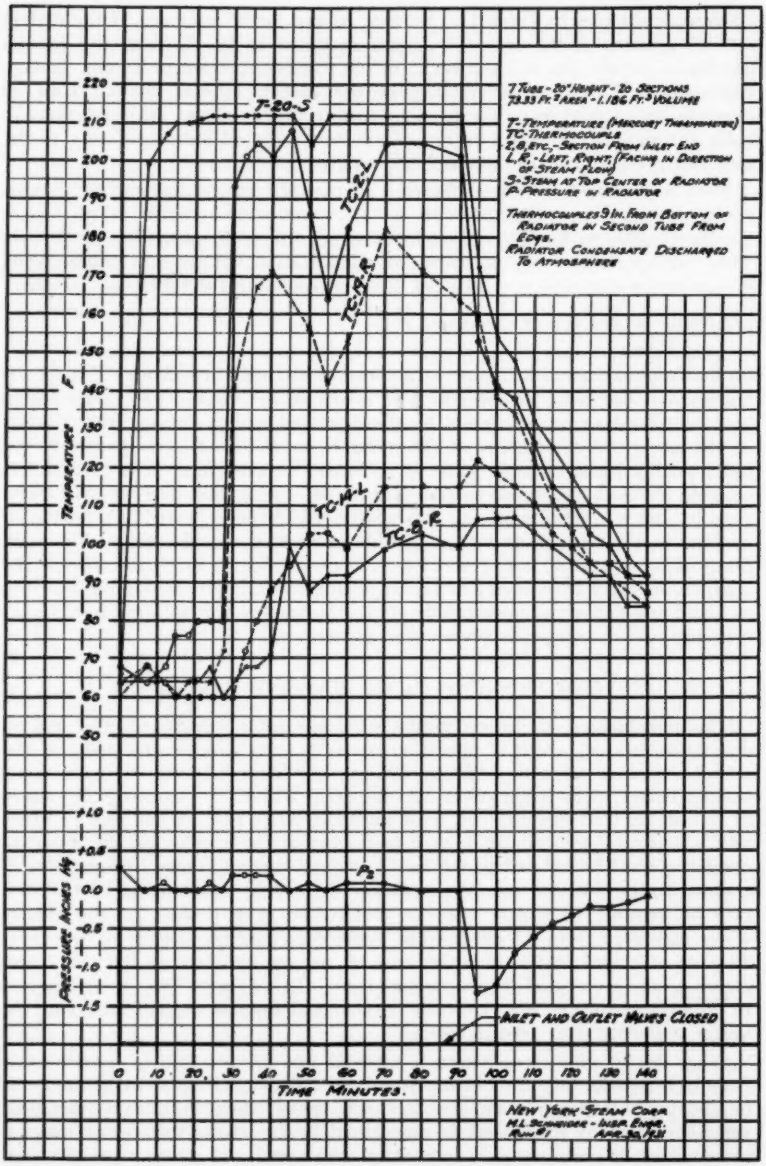


FIG. 2. DATA ON THE TEMPERATURES AND PRESSURES IN A PADIATOR DURING OPERATION

the solubility of the gases under these conditions is a function of their partial pressure in the gaseous phase, the answer to this question devolves totally on these solubility relations.

The authors believe that the use of a closed heater represents introduction of unnecessarily large amounts of gases into the steam, and should be avoided. On the basis of the analysis that follows, any of the other three types of heaters provides essential limitation of oxygen. The preference of the authors is for (1) or (2) above, since either, when operating properly, acts as a governor providing a limiting concentration of oxygen in the system, so far as the generating system is its source. Thus in the direct-control open heater, with ample venting by pure steam, the oxygen in the effluent water should be zero theoretically, but in reality will be as a customary maximum, 0.3 ml/liter, and on the whole will be less than this figure. Even this value, however, represents super-saturation, and once the water is vaporized, thus releasing all dissolved oxygen, any condensation of steam that occurs will contain but an extremely small amount of oxygen when saturated at the original partial pressure of oxygen involved, and the tendency will be to undersaturation rather than saturation, and oversaturation will be an impossibility.

The pressure in this second division of the system may vary from a usual maximum of 150 lb gage to 0 lb gage. At no point in the system, therefore, will there be tendency for the extraneous atmospheric gases to gain entrance into the system. Hence, definite figures can be obtained on the quantity of dissolved oxygen that can be present at saturation in any condensate that forms in contact with the flowing steam at these various pressures, because oxygen obeys the Law of Henry and Dalton, and the partial pressure of oxygen corresponding to any amount thereof present in the effluent water from the heater, or found by tests on condensed samples of the steam, is readily determined.

In Table 10-A calculations similar to those for carbon dioxide are given for arriving at the amount of oxygen at saturation in condensate in contact with steam at 150 lb and 0 lb gage pressure respectively, and for an oxygen content in the steam of 0.3 and 1.0 ml/liter. In Table 10, line 7, are presented the saturation concentrations of oxygen under these various conditions. From line 6 it may be noted that the partial pressures of oxygen in the steam are extremely small, and hence in the condensate are found very small saturation values of less than 0.0002 ml/liter in any of the cases illustrated.

This quantity (0.0002 ml/liter) of dissolved oxygen is extremely small. It means, therefore, that for all those internal surfaces of the system in which the partial pressure of oxygen in the flowing steam is uninterruptedly the controlling factor defining the quantity of dissolved oxygen, the quantity factor in dissolution of the metal is quite negligible, even though determination of oxygen in the accepted manner by condensation of the steam should show a total of 1 or 2 or even more ml/liter.

It would seem that one is prone to misinterpret the significance of this determination of total oxygen—to magnify in one's thoughts some thousand fold the possible corroding capacity of this condensate, because subconsciously one assumes that the condensate in the system in contact with the steam is the same as the dissolved oxygen concentration that is found in the specially-taken sample for total oxygen determination. Thus far, the discussion has been carefully limited to flowing steam, and it has been pointed out that this is not the case

TABLE 11. OXYGEN AND CARBON DIOXIDE IN THE STEAM—SECTION 1

Analyses of Steam or Condensate Drawn from Different Parts of the Steam System in Building A

(For conditions in the main steam lines on same dates, see Table 5)

No.	Date 1930	Where Taken	Steam or Condensate	Oxygen ml/l	Carbon Dioxide, ppm.	*pH* Values
1	Aug. 5	St. John *C* Meter	Steam	0.39	20.0	5.5
2	Aug. 6	Downstream side of valve 8 in. H.P. riser from 90-lb header	Steam	0.25	8.6	5.2
		St. John *C* Meter	Steam	0.34	0.0	5.2
3	Aug. 7	Trap at west end of 90-lb header	Steam	0.61	13.0	5.1
4	Aug. 8	St. John *C* Meter	Steam	0.39	18.5	5.2
		Steam Trap at *C* Meter	Steam	0.46	20.0	5.5
5	Aug. 12	Before trap on return from hot water tank 29th Floor	Steam probably mixed with considerable condensate	1.12	25.0	4.9
6	Aug. 13	5-lb header 29th floor	Steam	0.42	19.5	5.1
		Outlet of hot water tank trap, 29th floor	Condensate	1.10	6.0	5.8
7	Aug. 15	Return from trap on Coffee Urn No. 3 Kitchen Uptown Club	Condensate	0.35	22.3	5.1
8	Aug. 25	Inlet side of trap at heel of 8-in.	Steam	0.19	23.0	5.4
		Returns before vacuum pump	Condensate	4.48	2.0	6.6
9	Aug. 26	Returns on Pressure Gage System	Steam	0.7	39.0	5.8
10	April 21 Temp. of Condensate 56 F	St. John's Meter		0.35[a]	15.2[b]	5.3
11	April 21 Temp. of Condensate 60 F	19th floor		0.26[c]	16.4[d]	5.3

[a] Avg. of 6 tests, varying from 0.26 to 0.43 ml/l.
[b] Avg. of 5 tests, varying from 13.1 to 16.9 ppm.
[c] Avg. of 4 tests, varying from 0.21 to 0.34 ml/l.
[d] Avg. of 4 tests, varying from 15.3 to 16.4 ppm.

TABLE 11. OXYGEN AND CARBON DIOXIDE IN THE STEAM—SECTION 2

Analyses of Steam or Condensate Drawn from Different Parts of the Steam System in Building B

No.	Date 1930	Where Taken	Steam or Condensate	Oxygen ml/l	Carbon Dioxide, ppm.	*pH* Values
1	Nov. 23	Condensate E. Meter	Steam	0.43	25.9	5.1
2	Nov. 26	High Pressure drip from valet shop	Condensate	0.29	5.0	5.6
3	Nov. 25	Valet shop condensate at sub. meter	Steam	0.53	34.1	5.0
4	Nov. 25	Trap-outlet of presser valet shop	Condensate	0.27	3.0	5.3
		L.P. trap valet shop	Condensate below atmospheric press	1.43	5.2	5.1
5	Dec. 11	Grill room kitchen entering steam table	Steam	0.99	23.9	5.0
6	Dec. 3	Laundry—condensate from entrance to presser	Steam	0.75	22.9	5.0
		Laundry—condensate from entrance to mangle	Steam	0.49	24.0	5.0
7	Nov. 25	Laundry returns from presser before trap	Steam	0.58	28.3	4.9
		Laundry returns from mangle after trap	Condensate	1.50	20.1	4.9
		Laundry returns from mangle before trap	Steam	2.58	52.5	4.8
		Laundry return—basement—before pre-heater	Condensate	0.20	2.1	6.0
8	Dec. 1	Trap-inlet to hot water tank	Condensate	3.99	10.3	5.6
		Before trap inlet to hot water heater	Steam	1.10	15.8	5.1
9	Jan. 9, '31	Kitchen returns—after trap before manifold	Condensate	2.93	19.3	5.3
		Same. (See Notes.)	Condensate	1.21	10.6	5.6
10	Jan. 9, '31	Drip lines, H. P. risers before trap and manifold, under pressure. Sampled on left side	Condensate	0.79	None found	6.7
		Same. Sampled on right side, under pressure	Condensate	0.90	None found	6.7
		Attempt to take same during vacuum period	Condensate	3.40	None found	6.7

See explanatory notes, p. 150.

NOTES ON TABLE 11—SECTION 2

1. *Condensate E Meter.* Values are average of four determinations in each case.
2. *High Pressure Drip from Valet Shop.* Sample taken in sub-basement as 4th floor location was inaccessible. Values are average of determinations in each case.
3. *Valet Shop Condensate at Sub. Meter.* Sample could not be taken from trap at meter. Taken direct from meter. Values are average of three determinations in each case.
4. *Trap—Outlet of Presser Valet Shop.* Average of three determinations in each case.
   L. P. Trap, Valet Shop. Average of four determinations in each case.
5. *Grill Room Kitchen. Entering Steam Table.* Only permissible location in Grill room. Values are average of seven determinations on oxygen, six each on carbon dioxide and *pH* value.
6. *Laundry—Condensate from Entrance to Presser.* Average of three determinations in each case.
   Laundry—Condensate from Entrance to Mangle. Average of four determinations on oxygen, and three each on carbon dioxide and *pH* value.
7. *Laundry Return from Presser before Trap.* Average of four determinations in each case.
   Laundry Return from Mangle After Trap. Impossible to eliminate bubbles. Average of four determinations in each case.
   Laundry Return from Mangle Before Trap. Impossible to eliminate bubbles. Average of five determinations on oxygen and four each on carbon dioxide and *pH* value.
   Laundry Return—Basement—Before Preheater. Average of three determinations each.
8. *Trap—Inlet to Hot Water Heater.* Average of six determinations in each case.
   Before Trap—Inlet to Hot Water Heater. Average of four determinations in each case.
9. *Kitchen Returns—After Traps Before Manifold.* Vacuum pumps shut off, also line to small preheater and radiator by-pass. Average of two determinations in each case.
   Same. With above valves open. Water was rusty. Average of two determinations in each case.
10. *Drip Lines, H. P. Risers Before Trap and Manifold, Under Pressure, Samples on Left Side.* These are not hot water tank returns. *NOTE:* When steam is on, there is pressure in these lines, when steam is off, there is vacuum. This analysis represents the pressure period. Average of two determinations in each case.
    Same. Sampled on Right Side Under Pressure. Average of two determinations in each case.
    Attempt to take same during Vacuum Period. Flow discontinuous, 4 in. mercury vacuum. One sample each for oxygen and carbon dioxide, two for *pH* value.

since oxygen closely obeys the Law of Henry and Dalton. What will be true, however, where the steam is practically stagnant, and the condensate is discharged through a trap? In a radiator, for example?

Assume 0.3 ml/l. total oxygen in the incoming steam, and 0 lb gage pressure, *i.e.*, atmospheric pressure on the radiator. Condensate is discharged periodically to a trap, but no gases are vented from the radiator. Uniformity of temperature (212 F) and of composition of vapor throughout the radiator is postulated.

When final equilibrium is reached, the condensate will contain 0.3 ml/liter. That amount will be its final maximum, because that is just the amount of oxygen furnished by the incoming steam.

When this occurs, the partial pressure of oxygen in the vapor phase in the

TABLE 11. OXYGEN AND CARBON DIOXIDE IN THE STEAM

SECTION 3

Building C—(Report by New York Testing Laboratories)

No.	Date 1930	Where Taken	Oxygen ml/l.	Carbon Dioxide ppm.	*pH* Values
1	About April				
	Thursday	Incoming Steam	2.0[e]	18	5.5
	Friday	Incoming Steam	0.42[e]	15	...
	Saturday	Incoming Steam	0.30	18	...
	Monday	Incoming Steam	0.34	...	...
		50-lb High Pressure Line to Kitchen			
2	Friday	Direct Return	Nil	37	...
		Return Drip	0.05	5	...
	Saturday	Roof Garden Kitchen on Pressure Side	Nil	...	...
		Low Pressure Returns			
3	Friday	Pump Running	4.9	0.5	...
	Saturday	Pump Running	4.1	...	...
		Pump Not Running	3.1	...	...

[e] Average of three tests.

TABLE 11. OXYGEN AND CARBON DIOXIDE IN THE STEAM

SECTION 4

Building D (Hall Laboratories, Inc., Report)

No.	Date 1930	Where Taken	Oxygen ml/l.	Carbon Dioxide ppm.	*pH* Values
1	About January				
		North Heater	3.5	...	...
		before trap	3.2	...	...
		North Heater	1.4	...	...
		after trap	1.0	...	...
2		South Heater	...	...	...
		before trap	0.8	...	...
		South Heater	1.7	...	...
		after trap	0.5	...	...
3		St. John Meter	0.5[f]	...	...
4		At Kips Bay	0.9	...	...
5		Condensate Receiving Tank	...	...	7.0
6		Steam Corporation trap at inlet	...	...	8.2
7		Hot water Boiler Trap	...	...	7.3
8		Laundry Trap on Mangle	...	...	8.4
9		No. 6 Heating Meter	...	...	7.4

[f] Average of four tests varying from 0.2 to 0.7 ml/l.

radiator must be 0.265 lb or in other words, the vapor will be 1.76 per cent oxygen by volume, and since the original steam contains 0.000024 per cent, this value represents a concentration of more than 73000 fold. If one were to sample the vapor in the radiator as in the customary oxygen determination, taking care that no gaseous bubbles escape, he would find the oxygen concentration in the steam to be 22000 ml/liter.

Fig. 2 presents data on the temperatures and pressure in a radiator during operation. The radiator was of 20-in. height, 7 tubes, 20 sections, 73.33 sq ft area and 1.186 cu ft volume. The steam temperature T was measured by a mercury thermometer at top center of radiator; other temperatures TC were taken with thermocouples, inserted in sections 8 and 19, numbered from inlet end, and on the right side of one facing in direction of steam flow; and 2 and 14, on the left side. The pressure in the radiator was recorded in inches of mercury. The thermocouples were inserted 3 in. from the bottom of the radiator in second tube from edge.

The recorded steam temperature, once established, remained practically constant during the experiment, until the inlet and outlet valves of the radiator were closed, when it fell nearly uniformly with the other recorded temperatures. The temperatures as recorded by the thermocouples varied considerably at each position, and the temperatures of the several positions were different, and varied without any apparent relation to number of sections from inlet end.

The pressure was nearly constant until the inlet and outlet valves were closed, when it rapidly fell to —1.3 in. mercury, then over the next 45 min gradually rose to atmospheric pressure. This increase of pressure is very significant, as it denotes the readiness with which inleakage of air occurs. Had the system been tight, the pressure at the end, when the temperature was approximately 90 F would have been —28.5 in. of mercury plus whatever pressure there was from gases derived from the steam. The inleakage was more rapid, the larger the vacuum created by condensation of the steam, as shown by the decreasing slope of the pressure-time curve as atmospheric pressure was more nearly attained.

Condensation in the radiator may be assumed to be ¼ lb per hour per square foot of heating surface. Therefore condensation per hour was $\frac{1}{4} \times 73.33 = 18.33$ lb. Since each pound of water occupies 26.82 cu ft of space as saturated steam at 212 F, the 18.33 lb represents 492 cu ft of steam. Thus in the course of an hour, the condensation is $492 \div 1.186 = 415$ *radiatorsful*. In 24 hours therefore, 9,960 *radiatorsful* of steam condense.

On the supposition that composition of vapor is uniform throughout the radiator, that condensate is discharged periodically to a trap, but no gases are vented from the radiator—a supposition that results in higher partial pressure of oxygen in the radiator than actually exists—and that the incoming steam contains 0.3 ml/liter of oxygen, the oxygen concentration in the radiator at the close of the period is 0.23 per cent (approximately).

Condensate at 212 F in contact with this vapor contains at saturation 0.23 (per cent) $\times$ 17 (ml/liter solubility of oxygen at 212 F and partial pressure of 14.696 lb) = 0.039 ml/liter. At the temperature of approximately 100 F recorded by *TC-14-L* and *TC-8-R*, at which the solubility of oxygen is 23 ml/liter, the saturation value is 0.053 ml/liter. If the steam contained 1 ml/liter, in place of the 0.3 ml/liter assumed in these calculations, the concentration of oxygen in

the vapor at the close of a 24-hour period on the postulation of no loss during this period, would be approximately 0.77 per cent, and oxygen saturation in the condensate would be 0.13 ml/liter at 212 F and at 100 F, 0.18 ml/liter.

These figures are of interest because they demonstrate the relatively slight possible concentration of oxygen in the radiator, if its source is the steam. The low temperature of two of the thermocouples might of course be explained as due to the segregation of non-condensable gases (oxygen, nitrogen, carbon dioxide) in certain sections of the radiator, thus making possible a somewhat higher content of dissolved oxygen in the condensate having contact therewith. The fact that this occurs only in some sections, and that the temperature of condensate discharged at trap or sump is 140 — 170 F is proof that any such condition is not general nor permanent. Because of lack of sufficient turbulence in the confines of the radiator, the tendency is rather to obviate any concentration of gases by their removal. Thus the operation of the thermostatic valve for discharging condensate from the radiator is equivalent to blowdown on a boiler at the point of highest concentration of salts in the boiler water. The leak-valve on other radiators is comparable to continuous blowdown on the boiler. This biggest element in the oxygen-history of the radiator confines, however, is not the oxygen derived from the steam during the operating periods, but the composition of the vapors therein during the intermittent periods when the steam is turned off. As shown by the curves of Fig. 2, the vacuum developed in the radiator when the steam supply was shut off, was rapidly dissipated denoting ingress of air to take the place of the condensing steam. When the steam is again turned on, heating occurs first near the steam-entry, and gradually extends through the radiator, while meanwhile the atmospheric gases are driven before the steam, and discharge practically completely through the open trap or leak-valve. During the off-periods, the concentration of oxygen in the radiator bears no relationship to the oxygen in the steam, but is a function of the composition of the atmosphere.

It seems certain, therefore, that the maximum concentrations of oxygen in the condensate, as calculated above, are much higher than those that actually exist, and that even with a ml/liter of oxygen continuously present in the steam going to the radiator, the amount dissolved in the condensate from this source is too small to be of detriment to the surfaces the condensate contacts.

It is apparent that it is not a simple matter to obtain a condensate containing a few tenths ml/liter of oxygen if the source of supply of that oxygen is steam likewise containing only a few tenths ml/liter. There is a possibility that this might occur at totally dead ends, devoid of any periodic flow, and unvented. As for samples taken elsewhere, as from traps in any of the working parts of the system, any justification for ascribing several tenths ml/liter of oxygen therein to the presence of a similar total oxygen content in the flowing steam seems utterly impossible and absurd. The same considerations apply as well to carbon dioxide as to oxygen.

Discussion of Data from Various Buildings

In Table 5 are presented data on station steam in the distributing mains serving the buildings under consideration. Table 11 presents further data from different points. As an average value of oxygen in the steam as it leaves the

distributing mains of the generating system to enter the lines of the utilizing system, 0.5 ml/liter is generously high.

As shown in the preceding discussion, the partial pressure of oxygen in the steam, corresponding to this concentration of oxygen therein, is so low that the amount which can dissolve in any condensate is too slight to be considered a serious quantity factor in any corrosive action.

Discussion of returns before vacuum pumps is deferred to Division 3. At this point, however, it is pertinent to discuss the high oxygen values found in samples taken before the trap on the return from the hot water tank. These may be noted in Table 11 as follows: Sect. 1, Building *A,* Nos. 5 and 6; Sect. 2, Building *B,* No. 8; Sect. 4, Building *D,* Nos. 1 and 2.

In Building *A, steam* at the 5-lb header on the 29th floor contained 0.42 ml/liter of oxygen, 19.5 ppm. of carbon dioxide; *steam* before the trap on the return from the hot water tank contained 1.12 ml/liter of oxygen and 25 ppm. of carbon dioxide. On the other hand, the *condensate* from the outlet to the hot water tank trap contained 1.10 ml/liter of oxygen and 6.0 ppm. of carbon dioxide. The correctness of the analytical work in showing less carbon-dioxide content in the condensate is well illustrated by the *pH* values: When steam was condensed for sampling, using cooling coil, and getting into solution the 19.5 and 25 ppm. of carbon dioxide, the *pH* values were respectively 5.1 and 4.9; on the condensate, containing only 6.0 ppm. of carbon dioxide, the *pH* value was 5.8.

The steam in the hot water tank showed an increase in concentration of oxygen over that in the steam in the header of 2.7 fold; the carbon dioxide increase was 1.3 fold. This decrease in relative concentration of carbon dioxide leads to the suspicion that the increment of oxygen concentration was due in part to inleakage of air in which the ratio of oxygen to carbon dioxide is 500: 1, whereas in the steam the ratio is 1: 32.5. Suspicion changes to certainty when the dissolved gases in the condensate are considered. By the widest stretch of the imagination, steam containing 1.12 ml/liter of oxygen cannot form condensate at 140 — 170 F containing 1.10 ml/liter. On the basis of the Law of Henry and Dalton, saturation concentration is only 0.000016 ml/liter. Hence immense enrichment in oxygen must have occurred in the vapor phase contacting the sample taken from the trap.

Just how the condensate picks up its dissolved carbon dioxide is hard to explain. However, as noted heretofore, while the values of the carbon-dioxide determinations relative to each other are exact enough, the methods do not permit attaching too great significance to the absolute values.

Thus it is quite evident that all of the water coming to the traps from the hot water tank cannot contain one or more ml/liter of dissolved oxygen, if the source of that oxygen is the steam itself, because there is not that much oxygen associated with the steam. A part of the water might have this much oxygen, and the rest be relatively free from it. This interpretation of conditions, however, is negated by the fact that the samples taken from the outlet of the hot water tank trap show an amount of dissolved oxygen that is higher or as high as that in the steam. Doubtless, therefore, this larger supply of oxygen is obtained mainly not from the steam but from the inexhaustible source of the extraneous atmosphere.

Other examples that well illustrate these facts are found in Table 11, Section 1, Building *A*, No. 8; Section 2, Building *B*, Nos. 3 and 4, and Nos. 6 and 7; Section 4, Building *D*, Nos. 1 and 2, in which the samples before the trap were steam, those following the trap were condensate.

In the returns from the trap on the coffee urn No. 3 (kitchen) of the Uptown Club, Sect. 1, Building *A*, No. 7, only 0.35 ml/liter of oxygen is present. This fact calls attention to one factor in these pressure lines that must be remembered. At times it occurs that a shut-off valve on one of the pressure lines is located considerably back of the point at which the end of the pressure system ends, and the vacuum system begins. When this line is shut off, as in intermittent operation, then that section of the line between the shut-off valve and the normal beginning of the vacuum system in reality constitutes a part of the vacuum system and must be so considered in problems of corrosion.

In conclusion of the discussion on this division, attention is again directed to the boundaries set for it, which limits it to that part of the system, whether the distribution mains of the generating system, or the lines of the utilizing system, in which the steam pressure is always equal to or greater than the pressure of the surrounding atmosphere, and in which therefore flow of any leakage is from conduit into the atmosphere. There exists in this division no intermittency of steam pressure resulting from operating practices. Under these conditions, if the amount of oxygen and carbon dioxide associated with the steam as it leaves the generating system is limited to that characteristic of the effluent water from a direct contact vented heater pegged at 212 F, the partial pressure of these gases in the steam cannot possibly be sufficient to provide a concentration of either gas in any water condensed from the steam to provide either a deleterious quantity or rate factor of corrosion by said condensed water. In fact, the deductions from the Law of Henry and Dalton concede much greater leeway of oxygen content in steam than this without indicating it as causative of detrimental corrosive conditions in the utilizing system.

Problems of Utilization Division

In this division, characterized by intermittency of steam flow and pressure, and by inleakage of atmospheric gases, the origin of any deposits that are found can be adjudged with certainty when their chemical composition is known. If the origin is carry-over from the boiler, the deposit reflects the characteristics of the boiler-water sludge; if the origin is corrosion, the deposit is characteristically iron oxide, since iron is usually the only material available for corrosion, and in all cases is the material most susceptible to corrosion.

Analyses for Oxygen and Carbon Dioxide at Various Points in Building *A* and Building *B*

In the specific system under consideration, the data of Tables 5 and 11 have led to adoption of the figure 0.5 ml/liter as a generous estimate of the average amount of oxygen in the steam. Even if the average concentration had been 1.0 ml/liter the considerations set forth in the discussion of Division 2 show that condensate, not reinforced by oxygen from the atmosphere, would contain so little dissolved oxygen that its aid as the quantity factor in corrosion would be well-nigh negligible.

Both in Building *A* and in Building *B*, the data on oxygen presented in Table 11 show that excess thereof over and above that accounted for by the steam itself, is present in considerable quantity at different points. The latter building shows a worse condition than the former, the analysis of the returns being particularly indicative of much inleakage of air. In the data on Building *A*, considerable oxygen is found in the steam before the trap on the return from the 29th floor hot-water tank (Section 1, line 5), although the steam in the 5-lb header (line 6) contains only 0.42 ml/liter of oxygen. The *condensate* at the outlet of the trap contains 1.10 ml/liter. These data were discussed in the preceding sections of this report, and the conclusion was reached that in no conceivable way can the amount of oxygen in the condensate be derived from the oxygen in the steam. Table 11, Section 1, No. 8, shows that the returns before the vacuum pump contain 4.48 ml/liter of oxygen, and as the temperature of these returns lies within the limits of 140 — 170 F, this value is actually more than saturation for water at this temperature in contact with air, and blanketed with its own vapor. (See Table 1.)

In the samples taken at Building *B*, greater prevalence of high oxygen is found than in those taken at Building *A*. Some of this may of course come from condensation of steam with concomitant concentration of the non-condensable gases, as in the hot-water heaters, mangles and kitchen, but the concentration of oxygen so derived, as indicated for instance in Nos. 5, 6, 7 and 8 (Table 11, Section 2), can account for but slight dissolved oxygen in any condensate, in accordance with the conclusions arrived at in connection with Table 10. In No. 10 of Table 11, Section 2, is found definite evidence of the influx of air accompanying operation of the vacuum pump. The *pH* values of these samples are of interest—higher than those others noted in condensate further along in the utilizing system, and thus denoting the removal by this drip of the extremely small content of boiler water carried by the steam. The fact that no carbon dioxide is found in these samples is confirmatory of the view, previously expressed in part, that the absolute values of small quantities of carbon dioxide are very likely more a matter of indicators and end-points, than any reality. No analysis of the return water before the vacuum pump is available, but as the temperature thereof is 140 to 170 F as in Building *A*, the value is doubtless comparable to the 4.48 ml/liter found for Building *A*.

The composition of the vapor discharged by the vacuum pump has been determined by Markson in a typical case to be 0.7 per cent and 16.1 per cent by volume of carbon dioxide and oxygen respectively. More recent and extensive data, accumulated during the investigation of Building *A*, are presented in Table 12. These analyses show that the vented gas is largely air. Thus, in steam containing 0.5 ml/liter of oxygen and 19.6 ppm. of carbon dioxide, the ratio by weight of oxygen to carbon dioxide is 1 : 27.5; and in the air as analyzed in the basement was 148 : 1. In the vapors vented from the system the ratios are respectively; 38 : 1, 48 : 1, 144 : 1, and 147 : 1. Because of the similarity of these ratios in vented vapors to that in air, and because of their total dissimilarity to the same ratio in the steam, it is obvious that the oxygen and carbon dioxide brought to the system by the steam constitute a wholly negligible fraction of the vapors in the system. A simple calculation, based on steam containing 0.5 ml/liter of oxygen and 19.6 ppm. of carbon dioxide, emphasizes this conclusion. The total carbon dioxide, oxygen and nitrogen

in such steam amount to 226 cu ft at 170 F for each million pounds of steam. In Building *A*, the total heating surface in use (direct and indirect) is 120,272 sq ft. If condensation amounts to ¼ lb per square foot per hour, the steam requirement of the building is 30,068 lb per hour. Thus, if the vacuum pump removed all the gases supplied by the steam, it would vent only 6.8 cu ft per hour. This figure is a maximum, and negligible in comparison to the actual venting from the system as shown in the 6th column of Table 12, when the actual steam flow to the building was only 11,000 to 16,000 lb per hour.

Analyses of Deposits, Building A and Building B

In Table 13, Section 1, are given analyses of deposits from the Building *A;* in Table 13, 1-a, are found descriptions of the deposits and rough estimates of their quantity.

In viewing the table as a whole, the most impressive characteristic is that of line 11—*Magnetic*—defining with all certainty the origin of these deposits as of corrosive action. The quantity of iron oxide present expressed as ferric oxide (line 5) further shows that with two exceptions, Nos. 1 and 3, this substance dominates these deposits in large measure. From line 3 it is seen that the phosphate content—the contribution to these deposits of carry-over in the steam—is, in the main, practically negligible. Exceptions are Nos. 1, 1-a, 8 and 17.

TABLE 12. OXYGEN AND CARBON DIOXIDE IN GASES VENTED FROM THE HEATING SYSTEM OF BUILDING *A*

Tests made 4/23/31—*Analyses by Orsat*

Test No.	Time	Steam Flow (lb per hour)	High Pressure Header (lb per sq in. gage)	Air Temp. (Deg Fahr)	Cu Ft per Min Through Vents	Vacuum in Tank Inches Mercury	Gases per Cent by Volume	
							O_2	CO_2
1	7-8:07 A.M. ...	12,400	70	82	34.7	3.5*	21.0	0.4
2	11-12 N.	11,000	76	114	1.18	0	20.0	0.3
3	3-3:30 P.M. ...	15,300	69	110	3.22	0	20.2	0.1
4	3:30-4:00 P.M.	16,100	65	66	1.54	4.5	20.6	0.1
5	Air analysis in basement, 4/24/31						20.7	0.1

* Vacuum pump running at start of test. Pump stopped at 8:00 A. M.

The difference between Nos. 1 and 1-a is probably due to variations in operating practice at different seasons of the year. Thus No. 1 was taken in the winter, when the rating on the boilers in the generating system and hence the carryover therefrom was maximum. No. 1-a was taken in the summer, when rating and carryover are a minimum, and when, therefore, without any increase in rate of corrosion any material of corrosive origin would constitute a greater part of the sample. The elimination by the generating system of moisture from the steam is now so nearly complete, that all samples of deposits from the 90-lb headers will doubtless conform more nearly to 1-a than to 1.

The high silica in No. 3 is the cause for the lower value of iron oxide. The fact that this trap had never been cleaned since the installation of the system

TABLE 13. DEPOSITS TAKEN FROM VARIOUS SOURCES—BUILDING *A*

Analyses in Per Cent

SECTION 1

	1	1A	2	3	4	5	6
	Traps on 90-lb Header	Traps on 90-lb Header	Trap at High Pressure 60-lb Line, 16th Fl. Water Heater	Low Pressure Drip Trap Inlet of Water Heater 16th Fl.	Trap at Outlet from Water Heater 16th Fl.	Primary Reducing Valve 19th Fl.	12-in. Reducing Valve 19th Fl.
1. Sulphur Trioxide (SO_3)	1.4	2.5	1.3	1.4	1.7	4.0	Not enough sample
2. Carbon Dioxide (CO_2)	Included in Ignition Loss.						
3. Phosphorous Pentoxide (P_2O_5)	16.6	9.8	1.3	2.0	1.7	2.5	2.0
4. Silica (SiO_2)	22.3	5.9	19.0	36.5	12.7	3.7	3.6
5. Ferric Oxide (Fe_2O_3)	28.7 (items 5 and 6 together)	80.1	76.7	41.6	70.7	76.7	64.1
6. Aluminum Oxide (Al_2O_3)		4.8	6.8	10.5	9.7	9.6	Not enough sample
7. Calcium Oxide (CaO)	15.0	8.1	0.7	6.0	6.0	2.6	8.8
8. Magnesium Oxide (MgO)	11.9	1.1	1.1	1.7	0.7	0.7	5.4
9. Loss at 105 C	0.5	0.1	0.4	0.4	0.4	1.0	2.7
10. Ignition Loss	3.0	0.2	0.1	2.3	4.9	8.7	10.6
11.	slightly magnetic	magnetic	magnetic	magnetic	magnetic	magnetic	magnetic

1. *Trap on 90-lb Header.* (Sample taken in spring.) No record of quantity present.

1A. *Traps on 90-lb Header.* (Sample taken in summer.) 1 in. bucket trap, light deposit in East trap, fairly heavy in West. Deposit caked. Magnetic, dark gray powder, part was soft, partly gritty. East trap cleaned in June, West in April. Amount: East, 1-2 gms. (cleaned 1 mo. previous.) West, 50 gms. (cleaned 3 mo. previous).

2. *Trap at High Pressure, 60 lb-Line, 16th Floor, Water Heater.* Bucket trap. Small amount of deposit loose in trap. Magnetic, sandy mixture of white, black and brown particles. Trap had never been cleaned. Amount, 30-35 gms.

3. *Low Pressure Drip Trap, Inlet of Water Heater, 16th Floor.* Ball trap. Deposit fairly loose, about 1/64 inch thick. Magnetic, brownish powder, mostly soft with sandy particles. Trap had never been cleaned. Amount, 30-40 gms.

4. *Trap at Outlet from Water Heater, 16th Floor.* Ball trap. Deposit about 1/64 in. thick, caked. Magnetic, brown, soft powder. Trap had never been cleaned. Amount, 30-40 gms.

5. *Primary Reducing Valve, 19th Floor.* Deposit about 1/32-1/16 in. thick. Magnetic, orange-brown, soft powder. Cleaned in February. Amount, 5-10 gms.

6. *12-Inch Reducing Valve, 19th Floor.* Deposit was paper thin. Some pitting on cast iron housing and upper cap of valve body. Magnetic, brown, soft powder. Cleaned in March. Amount, 15-20 gms.

TABLE 13. DEPOSITS TAKEN FROM VARIOUS SOURCES—BUILDING *A*

Analyses in Per Cent

SECTION 1 (Cont'd.)

	7	8	9	10	11	12	13	14
	Drip Trap Inlet to Water Heater 29th Fl. (Low Pressure)	Traps on 53rd Fl.	Pipe of Return Line of Pressure Gage System	Dirt Pockets Pressure Gage System House Heating System	Dirt Pocket Pressure Gage System 26th Fl. Water Heater	Dirt Pocket Pressure Gage System 29th Fl. Water Heater	Drip of Pressure Gage System Trap 29th Fl. Hot Water System Heater	Drip on Pressure Gage System 16th Fl. Water Heater
1. Sulphur Trioxide (SO_3)	2.9	1.4	1.2	2.6	2.6	2.3	0.8	1.7
2. Carbon Dioxide (CO_2)	Included in Ignition Loss.							
3. Phosphorous Pentoxide (P_2O_5)	2.3	12.0	2.1	1.0	1.7	1.8	0.5	0.6
4. Silica (SiO_2)	4.8	13.8	0.8	2.6	4.2	3.2	0.9	1.1
5. Ferric Oxide (Fe_2O_3)	93.0	61.1	86.4	61.1	80.1	81.8	92.4	88.3
6. Aluminum Oxide (Al_2O_3)	5.7	4.7	5.8	15.2	8.7	4.8	3.5	1.8
7. Calcium Oxide (CaO)	3.7	12.0	1.0	5.6	0.5	3.5	1.7	2.7
8. Magnesium Oxide (MgO)	0.9	0.5	0.8	2.3	1.5	1.5	0.1	1.0
9. Loss at 105 C	0.3	1.2	0.2	1.4	0.2	0.4	0.8	2.2
10. Ignition Loss	2.0	2.9	7.7	11.1	7.7	8.2	1.9	6.2
11.	magnetic	magnetic	magnetic	magnetic	magnetic	magnetic	magnetic	magnetic

7. *Drip Trap Inlet to Water Heater, 29th Floor.* (Low pressure.) Ball trap. Deposit same in appearance and quantity as that on 16th floor. Magnetic, black, soft powder. Never cleaned.
8. *Traps on 53rd Floor.* Ball trap. Deposit was soft. Magnetic, brownish, gray, soft powder. Never cleaned. Amount, over 100 gms.
9. *Pipe of Return Line of Pressure Gage System.* One inch pipe practically closed with brown deposit. Magnetic, reddish-brown, soft powder.
10. *Dirt Pockets Pressure Gage System House Heating System.* Magnetic, brown and orange-brown particles, soft, easily powdered. Cleaned in spring. Amount, 10 gm.
11. *Dirt Pocket Pressure Gage System, 26th Floor Water Heater.* Magnetic, dark brown, scaly, soft powder. Cleaned in May 2 mos. previous to this examination. Amount, 5 gms.
12. *Dirt Pocket Pressure Gage System, 29th Floor Water Heater.* Magnetic, scaly, dark brown powder. Cleaned in May 2 mos. previous to this examination. Amount, 5-10 gms.
13. *Drip of Pressure Gage System, Trap, 29th Floor, Hot Water System Heater.* Magnetic, reddish-brown, soft powder. Amount, 10 gms.
14. *Drip of Pressure Gage System, 16th Floor Water Heater.* Magnetic, brown, soft powder. Amount, 3-5 gms.

TABLE 13. DEPOSITS TAKEN FROM VARIOUS SOURCES—BUILDING *A*

Analyses in Per Cent

SECTION 1 (Cont'd.)

		15	16	17	18	19	20	21	22
		High Pressure Trap 19th Fl.	Heel of 8-in. Riser 90-lb Line	Inside of Body of 12-in. Reducing Valve 19th Fl.	Uptown Club Kitchen 26th Fl. Coffee Urn Returns	High Pressure Trap 53rd Fl.	Thermostat Trap Low Pressure Header, 19th Fl.	Deposit on Steel Nipple Following Radiator	Deposit on Brass Nipple Following Radiator
1.	Sulphur Trioxide (SO_3)	0.6	nil	...	...	...	...	nil	nil
2.	Carbon Dioxide (CO_2)	...	...	...	...	...	...	...	...
3.	Phosphorous Pentoxide (P_2O_5)	1.4	2.1	19.8	5.4	1.0	1.3	trace	trace
4.	Silica (SiO_2)	15.5	7.1	...	...	...	...	0.7	0.3
5.	Ferric Oxide (Fe_2O_3)	78.7	95.2	...	...	97.3	...	91.1	92.0
6.	Aluminum Oxide (Al_2O_3)	3.0	...	...	...	...	...	...	...
7.	Calcium Oxide (CaO)	2.9	trace	...	...	...	...	nil	nil
8.	Magnesium Oxide (MgO)	1.2	trace	...	...	...	...	nil	nil
9.	Loss at 105 C	1.5	...	...	...	...	...	...	...
10.	Ignition Loss	1.3	...	...	...	...	...	8.8	7.9
11.	....	magnetic	magnetic	slightly magnetic	magnetic	magnetic	magnetic	magnetic	slightly magnetic

15. *High Pressure Trap, 19th Floor.* Bucket trap, deposit about 1/64 in. thick, caked. Black, magnetic, soft powder, gritty. This trap never cleaned. Amount, 10-15 gm.
16. *Heel of 8-in. Riser, 90-lb Line.* Magnetic, dark gray to black, gritty grains, caked.
17. *Inside of Body of 12-in. Reducing Valve, 19th Floor.* Slightly magnetic, light color.
18. *Uptown Club Kitchen, 26th Floor, Coffee Urn Returns.* Black, magnetic, soft powder.
19. *High Pressure Trap, 53rd Floor.* Brown, soft, magnetic powder, suspended in water.
20. *Thermostat Trap Low Pressure Header. 19th Floor.* Magnetic, reddish-brown, flaky.
Samples 16-20 were too small for complete analysis.
21. *Deposit on Steel Nipple Following Radiator.* Brown deposit, readily pulverized. Magnetic.
22. *Deposit on Brass Nipple Following Radiator.* Brown deposit, readily pulverized. Magnetic.

probably accounts for this value. This accounts likewise for the high silica of Nos. 2, 4, 8 and 15.

The high phosphate of Nos. 1 and 1-A indicate that as carryover occurs, much of the phosphate material is eliminated at the 90-lb header. That of Nos. 8 and 17 probably dates back considerably, as these traps had never been cleaned, and the quantity of carryover from the generating station has been progressively growing less. No explanation is apparent why phosphate should be low in No. 2, which is a sample from a trap on the high pressure line that had never been cleaned, especially in view of the evidence afforded by the high *pH* values of condensate (Table 11, Section 2, No. 10), that removal of any carryover of boiler water occurs in this part of the system.

The further measure of the separator installed just prior to the distributing mains represents an additional effective barrier against any quantity of deposit arriving at the 90-lb header, and thus will eliminate or minimize any troubles caused thereby, either at the main reducing valve located at this point, or at those other points noted as characterized by high phosphate in this series of tests. Under these conditions, it seems likely that the separator installed in Building *A* will have little or no work to do, thus rendering its installation unnecessary.

In Table 13, Section 2, are presented results on various samples taken from Building *B*. Complete analyses were not made, but the descriptions *Magnetic, trace of phosphate* or *no phosphate* and *oxides of iron* show that these deposits belong in the category of corrosion products, just as truly as those from Building *A* which have just been discussed.

The corrosion which these buildings have experienced requires real quantities of oxygen. The corrosive possibilities of the present steam with its 0.5 ml/liter of oxygen are practically negligible. No relief from the deposits so excessively high in iron can be expected to result from any further measures the generating system may take either to eliminate carryover or to reduce the oxygen concentrations in the steam. All considerations that have been discussed point to minimal effects from considerably higher oxygen concentration in the steam than have existed heretofore, in comparison with the effects induced by the hundred fold times this amount of oxygen that intrudes itself into the system by sundry paths.

It is the opinion of the authors that these deposits are products of corrosion caused by the large quantities of inleakage oxygen, and that their elimination or satisfactory limitation will require elimination of such inleakage or/and establishment of these conditions in the system which will minimize the corrosive action of any oxygen which does not get into the system.

Analyses 21 and 22 of Table 13, Section 1, are of particular interest. In this case, practically identical corrosion products, formed in identical loci on different radiators, are found, one on a steel and one on a brass nipple. In each case, they filled the ½-in. nipple, save for an opening of about ⅛-in. diameter at the center. The surfaces of the deposits were marked by convolutions as though the condensate, discharged from the radiator through trap, passed through the nipples with a swirling motion.

The two deposits were different in one respect. In the steel nipple the deposit close to the metal surface was strongly magnetic; closer to the surface

of the deposit and nearer the center of the nipple it was only slightly magnetic. The entirety of the deposit on the brass nipple was only slightly magnetic.

The body of the trap between nipple and radiator is apparently of corrosion-resistant steel; the thermostatic element of ordinary steel, copper plated. Corrosion deposit covers both, though to no degree of thickness commensurate with that of the nipples.

One definite conclusion may be reached, namely, that the dissolution of metal occurs at other surfaces than those on which deposition eventuates. This must be so, else the deposit would not be found on the brass nipples.

Where does the dissolution of metal occur? What is the mechanism of formation of the deposit in the nipples?

Before attempt is made to answer these questions, it will be well to consider

TABLE 13. DEPOSITS TAKEN FROM VARIOUS SOURCES—BUILDING *B*

SECTION 2

Specimen Marked	Description of Sample	Analysis
Cafeteria inlet to trap.	Reddish-brown, adherent, magnetic.	Oxides of iron, especially Fe_3O_4; no phosphate.
Grill-steam table. Exp. trap outlet.	Dark brown, adherent, magnetic.	Oxides of iron. No phosphate.
Cafeteria-steam table trap outlet.	Dark brown with light brown spots. Adherent, magnetic.	Oxides of iron. No phosphate.
Cafeteria and Grill-room common return.	Dark gray to black. Adherent, magnetic.	Oxides of iron. Trace phosphate.
Grill-room steam table. Exp. trap inlet.	Black to yellowish-brown. Flaky, easily chipped, magnetic.	Oxides of iron. Trace phosphate.
Laundry. Common return from Eagle press.	Reddish-brown, adherent, magnetic.	Oxides of iron. No phosphate.
Laundry. Am. Eagle Press exp. trap inlet.	Reddish-brown, adherent, magnetic.	Oxides of iron. No phosphate.
Laundry. Am. Eagle Press. Exp. trap outlet.	Reddish-brown, adherent, magnetic.	Oxides of iron. Trace phosphate.
Hot water tank return. Trap outlet nipple.	Dark brown, adherent, magnetic.	Oxides of iron. Trace phosphate.
Hot water tank. Trap inlet nipple.	Dark brown, adherent, magnetic.	Oxides of iron. Very faint trace of phosphate.

the data presented in Table 14, which show that the rapid development of these deposits in the period of a few months is specific to Building *A*. Thus, whereas in Building *A* the deposit builds quickly on the nipples and elbow (Nos. 1 — 3) following the radiator discharge trap to 0.00075 — 0.0018 lb per square inch of surface exposed, in Building *E* (No. 4), it reaches only 0.00046 lb per square inch in seven years. Furthermore, in the latter case, the deposit is very hard and dense, and restricts but very slightly the diameter of the tube. Because of the little nodules apparent over the surface, which are black beneath the surface covering, and because of its extremely tight adherence to the metal, it is the authors' belief that the latter deposit represents formation *in situ*, and once definitely established, forms a fairly impermeable coating which resists further attack. The deposit in the nipples from Building *A* is soft, devoid of nodules, and much more loosely attached, indicative of the special conditions

surrounding its accumulation. To answer the above questions, these special conditions must be recognized. When the radiator is functioning as a heating unit, its surfaces are wetted by water in contact with a gaseous phase of very low partial pressure of oxygen and carbon dioxide. Hence both the capacity and the rate factor for dissolution of the metal are small. After the steam in the radiator system has been turned off, the temperature drops quickly as shown by the curves of Fig. 2. Condensation should occur as quickly, and before the partial pressure of oxygen has increased to sizable value because of inleakage of air, the surfaces of the radiators should have drained thoroughly, and without leaving casual droplets of water adhering thereto. This last must be the case, since one of the best tools of the chemist for obtaining clean surfaces which drain uniformly is to subject them to steam condensation for a few hours, whereby all grease films are removed. Such drainage doubtless quickly results in dryness of the radiator surfaces, and hence immunity from the quantity

TABLE 14. DEPOSITS ON NIPPLES FOLLOWING TRAP OUTLET OF RADIATOR

Nos. 1-3 *from Building A: No.* 4 *from Building E*

	1	2	3	4
	Nipple to Trap	Elbow	Nipple Following Trap	Comparison Nipple
Length, inches	2.64	[a]	2.43	2.04
Inside Diameter, inches	0.51	0.91	0.52	0.58
Inside Area, sq in.	4.23	1.4	3.98	3.71
Wt. Deposit, lb	0.0036	0.0025	0.0030	0.0017
Wt. Deposit, lb per sq in.	0.00085	0.0018	0.00075	0.00046
Character of Deposit	Soft	Soft	Soft	Hard
Period of Formation	A few months	A few months	A few months	Seven years

[a] Long radius 2.76 in.
Short radius 1.75 in.
Sector 0.75 in.

dissolution that would ensue with certainty, were they wet when the partial pressure of oxygen became high.

In Building *A,* however, all heating returns from the various heating levels feed into a common vacuum pump inlet header. The hot water tank returns feed into the heating return lines connected to the system from the 32nd to 55th floor. The 19th floor low pressure heating header discharges into the intermediate and low heating system returns, i.e., below the 32nd floor, by thermostatic drip traps. These thermostatic traps are opened when the system is cooling, since a vacuum is created on the inlet side by reason of condensation of the steam in the low pressure header and in the heating system. Air can enter the hot water trap returns by vacuum breakers, which are at present set for high enough vacuum to function merely as a protective device and, under normal hot water tank operation, are inoperative; and by air vents on the ball float hot water tank traps—a change on these vents to one-way vacuum-type vents would preclude the intake of air at this point. The pressure gage line from header at the 55th floor heating, 19th floor heating, 29th floor hot water and 19th floor hot water are thermostatically trapped in the pump room in the basement and discharge into the sump through a common return line. These

traps are open when the steam headers to which they are connected are shut down, and when a vacuum is created in them by condensation, suction occurs back through the pressure gage lines and traps. A check valve has been installed on the discharge line to prevent this condition.

At various points in the system, therefore, air inleakage occurs, immediately after the steam in the radiator system has been turned off. The hot water systems, which operate well into the night—in fact, carry their heaviest load then, due to cleaning operations—discharge into the heating return lines and therefore keep them supplied with hot moist vapors that will provide condensate as they contact cooler surfaces, as a radiator, by being drawn therein by vacuum conditions.

In Building *A,* it is known that the vacuum obtaining shortly after the steam in the radiator system is turned off disappears entirely after a few hours. Thus partial pressure of oxygen in the radiators becomes very large. If the surfaces were dry, any resulting damage would be small, a conclusion fortified by the thousands of radiators that have not given trouble over a period of years. But specifically in Building *A,* connections are so arranged that presumably moisture may condense on the radiator surfaces in the presence of a considerable partial pressure of oxygen. Hence, while the rate factor of corrosion, as typified by relatively large carbon-dioxide content in the condensate is not high, the quantity factor is unlimited.

It is possible to retard so ready inleakage of oxygen into the system, as indicated previously; it is probably impossible to wholly prevent it. On the other hand, the possibility of hot moist vapor being drawn back into the radiators can be minimized. In both hot water tank rooms there are high pressure drip traps dripping the supply lines from the 90-lb riser to the hot water tank. These traps in turn discharge into a separate so-called high pressure return line which in turn, discharges into the sump, and is independent of the heating system. The pressure in this line is atmospheric at all times. The hot water tank returns can readily be discharged into the high pressure return line, and thus completely isolate the hot water tank returns from the heating system.

The deposits that are found in the nipples therefore have their origin in corrosion of the radiator surfaces. Iron dissolves from these surfaces as ferrous hydroxide, is oxidized to ferric hydroxide, precipitates, and when the water is discharged, is carried along and finds a resting place on the limited surfaces of the nipples. This explanation is in agreement with the fact that the deposits are softer and less dense than those formed *in situ,* and likens the formation of the convolutions on the surface of the deposit to the ripple marks in the sand of the shore. It is interesting to note that after Mr. Finnegan had sawed open a radiator and examined the internal surfaces, he wrote as follows: "The radiator is quite clean, and it is doubtful if enough deposit for analysis can be obtained from it." As the quantity of sample required for analysis is 5 grams or less (less than ⅙ ounce), it is apparent that deposit adherent to the surfaces is well-nigh nil. This is unusual, and is probably the result of the unusual cleanliness of the radiator surfaces, under which condition any moisture is in the form of films and not casual droplets. The absence of pitting on the interior surfaces of the radiator is evidence that dissolution has been general over them, and not merely at points at which

droplets of water by collecting expedited union of any dissolved metal with oxygen derived from the contacting vapor.

SUMMARY AND CONCLUSIONS

For an intrinsic realization of the factors that control the relations of steam to metal in steam heating systems, certain of the fundamental laws governing solubility in water, and reactions therein must be understood. The earlier part of the report, therefore, has been devoted to consideration of some of these laws, while in the latter part, their application to the specific problems of a steam-generating system and some of its utilizing systems has been pointed out. Both generating and utilizing systems are typical of the modern steam heating industry, hence the conclusions drawn on the data are of general applicability.

When the report was undertaken, it was thought that two general types of problems were involved: (1) formation at various points in the utilizing system of troublesome deposits, the origin of which was primarily carryover of boiler water and sludge by the steam; (2) corrosive action, and formation of deposits thereby in the utilizing system, with the question to be answered of what responsibility therefore should be borne by the steam furnished by the generating system.

Particularly at Building *A*, the formation of deposits in the utilizing system had been troublesome and had been ascribed to transportation of boiler sludge by the steam. Some months ago, when the inspection of one of the main gate-valves at the 90-lb header showed it to be practically devoid of such deposit, it was suspected that deposits throughout the system had as their origin the metal of the system degraded into such deposits by conditions inherent in the system itself—in other words, that *corrosion*, and not carry-over in the steam, constituted the problems at Building *A*, as elsewhere. The correctness of this suspicion is demonstrated by the analyses of deposits presented in Table 13, which show that all the samples are very high in, and many consist almost solely of, iron oxide.

Therefore, while some general consideration has been given to the elimination of carry-over in the steam, and to the means adopted therefor in this instance by the generating system, the main problem in all cases pertinent to this report has been corrosion.

According to the generally accepted view, corrosion in all cases comprises a *velocity or intensity factor*, and a *quantity factor*. The velocity factor is a function of the *pH* value of water, since rate of dissolution of iron is dependent on this. When the water is condensate derived from steam which was generated from alkaline boiler water, the *pH* value is dependent practically upon the amount of carbon dioxide associated with the steam. The quantity factor is a function of available *dissolved* oxygen, and hence of the concentration and actually available quantity of oxygen in the contacting vapors. From the standpoint of elimination of corrosion, the ideal would be zero intensity and quantity factors; as this is an impossibility in practice, it is necessary to establish for both factors tolerances that provide sufficient limitation thereof to render any corrosive action inappreciable in amount.

The two sources of oxygen and carbon dioxide are the steam and the atmos-

phere. In the steam, as illustrated at Kips Bay, carbon dioxide predominates in the ratio of 27.5 parts to 1 part of oxygen by weight. In the atmosphere, oxygen predominates in the ratio of 500 parts to 1 part of carbon dioxide by weight in pure air, and in the ratio of 148 to 1 in the basement of Building *A*. In general, therefore, in Divisions 1 and 2 of the steam heating system, in which no inleakage of atmospheric gases occurs, and in Division 3 as well, so long as inleakage does not occur, the intensity factor of corrosion, as typified by carbon dioxide, predominates in any action on the metal; when inleakage occurs in Division 3, the quantity factor, as typified by oxygen, immediately becomes the mainspring in any corrosion therein, owing to its huge preponderance over carbon dioxide in the atmosphere.

As a limiting concentration of carbon dioxide, the value of 15-20 ppm. has been mentioned as satisfactory. In pure condensate formed from steam of the carbon-dioxide content characteristic at Kips Bay (19.6 ppm.) and unbuffered by any dissolved salts, *pH* values respond by large changes to the slightest traces of alkalinity, and hence any capacity for dissolution of metal is lacking. In fact, in view of the conclusions reached on the basis of the Law of Henry and Dalton, it would seem that greater tolerance would be harmless. On the other hand, if condensation were to occur in dead ends, or at other points, without provision for regulated discharge of any non-condensable gases, the attendant largely increased concentration of carbon dioxide, and at times, of oxygen also, might give rise to conditions favorable to rapid corrosion in quantity. Therefore, since the value of 15-20 ppm. or less of carbon dioxide in the steam is obtained with no difficulty, if sagacious choice is made of a method for processing the feed water, we believe that this value represents a satisfactory tolerance.

In obtaining the concentration of carbon dioxide and of oxygen in the steam, the usual procedure is to condense a sample of steam by passing it through a cooling coil, care being taken that all gases are dissolved in the cool condensate. The determinations for carbon dioxide, oxygen, and *pH* value are then made on this sample. The values so obtained represent truly the composition of the *steam*, but have frequently been regarded as also applicable to *condensate formed from and in contact with the steam*. This is not the case, and the values so obtained do not apply to such condensate. Thus, the *pH* value in samples of condensed steam from Kips Bay Station averaged 5.1, and the carbon dioxide averaged 19.6 ppm.; at the same time, samples of condensate had an average *pH* value of 5.8, and a carbon-dioxide content of 7.6 ppm. These differences occur, because in steam such as that from Kips Bay, the partial pressure of carbon dioxide and of oxygen therein is extremely small.

In any consideration of the relations of steam and metal, therefore, due care must be exercised at all times that proper differentiation be made between the characteristics of the steam itself, and the condensate formed therefrom and in contact therewith.

Probably the greatest need for this discretion arises in connection with appraisals of the relation of oxygen content of the steam to corrosion occurring in the utilizing system. Thus it has frequently happened that steam has been sampled and condensed in a cooling coil as noted above, the oxygen content of the cooled condensate determined, and this value assumed to represent the amount of dissolved oxygen in any condensate formed from and in contact

with the steam. The assumption is absurd, because the Law of Henry and Dalton states that the concentration of dissolved oxygen is proportional to the partial pressure of the contacting gaseous oxygen. As a matter of fact, only a very small fraction of the value obtained by the determination on condensed steam will dissolve in condensate formed in contact with the steam, and only in dissolved form is oxygen effective in promoting corrosion.

Limitation of oxygen to the tolerance permissible in the steam is a function of the feed water heater. As previously noted, the lower limit obtainable with deaerating heater is approximately 0.025 ml/liter. This last trace may be removed if desired by chemical fixation following the heater. There is no particular advantage in such complete limitation however, so far as Divisions 1 and 2 of the system are concerned, since ever 2 or 3 ml/liter of oxygen in the steam exerts so slight partial pressure, that the quantity factor of corrosion of any condensate in these two divisions is inappreciable. Neither is there any advantage, so far as Division 3 is concerned, until the exclusion of atmospheric oxygen therefrom is made certain. Even at points of condensation under stagnant conditions as in radiators, accumulation of oxygen having its source in the steam, to sufficient partial pressure to provide in the condensate any sizable fraction of its concentration in the steam, is obviated by the blowdown of the vapors in the radiator at times of trap discharge, or charge, or through leak-valves. Besides, these amounts of oxygen are slight in comparison with those in the radiator during off-periods.

One can therefore find no fault with the 0.5 ml/liter of oxygen characterizing the steam furnished Buildings *A* and *B* by the Kips Bay Station. It is true that the condensate drawn at different points in these buildings in Division 3 of the system quite generally was found to have an oxygen concentration of 1 or 2 ml/liter, or even higher; but, in the light of the laws of partial pressure, and of data such as those in Table 12, it is obvious that this concentration of dissolved oxygen is originating not from partial pressures of gaseous oxygen derived from the steam, but from the intrusion into the system of the surrounding atmosphere with its infinitely greater proportion of oxygen.

The responsibility for oxygen so derived rests not with the steam generating system, but with the physical characteristics of the system in the building, such as its type and design, and the quality of workmanship in its assembly, and with the operating practice.

Manifestly, steam of a requisite quality, as defined in the preceding pages, is guilty of no part in the corrosion activities induced by intrusion of atmospheric oxygen into Division 3 of the system. There is therefore much leeway in choice of the upper limit of tolerance for oxygen. Not from necessity, but from preference, and a desire to be far within any questionable limits, we believe the upper limit of tolerance for oxygen might well be placed at 0.3—0.5 ml/liter, a value readily and assuredly attained.

If, by design of the system, care in its assembly, and choice of operating procedure, the oxygen concentration in Division 3 of the system could be restricted to that furnished by steam of the quality specified in this report, all problems of corrosion would become of vanishingly small consequence. Although the present-day technique of the well-operated steam generating station could readily and with certainty reduce the concentration of oxygen and carbon dioxide to a tithe of the value specified, to do so is of no practical value so

long as the steam passing from Division 2 of the system into Division 3 is degraded in quality by influx of oxygen from the inexhaustible reservoir of the atmosphere, and in amount dependent upon the physical and the operating characteristics of Division 3.

It is in Division 3, or the utilizing system, that any problems of corrosion are apt to become acute. From the very nature of Division 3, it is too much to expect that *all* inleakage of air thereinto can be prevented. Although such inleakage may and does occur in all utilizing systems, in the great majority of cases corrosion is so slight as to be of no moment. When corrosion does occur, it is usually restricted to well-defined areas in the system. It is therefore logical to assume that special conditions exist at such areas for the remedy of which individual measures are required.

In the quest for remedial measures for these cases, it is pertinent to inquire whether district heating steam or steam generated in the individual boiler plant in each building offers any advantages the one over the other in obviating possibility of corrosion. The answer is brief and entirely definite: *The maintenance of the requisite qualifications in the steam, not its source, is the telling factor.*

In the district heating plant, the make-up water is usually large in amount, little condensate being returned to the plant; in the individual plant, the condensate is returned, so that little make-up is required. In either case, the water requires treatment to preserve the evaporative surfaces of the steam generator in satisfactory condition; the same deaeration is required in either case, for limitation of oxygen in the steam is dependent, not on the amount of oxygen in the water entering, but in that leaving the heater; the same precautions are requisite in either case to prevent carry-over in the steam. The same permissible tolerances of carbon dioxide, oxygen and pH values in the steam obtain in one case as in the other. Continuously meeting these specifications, the steam, whatever its source, cannot be held responsible for the development of corrosion in the utilizing system.

It would be egregious to suppose that avoidance or solution of these problems of corrosion resides in mere substitution of an individual boiler plant in the building itself in place of the district heating steam. Rather, in so doing, to the problems that are his in any event, the consumer adds that of obtaining the skilled and understanding operation in his boiler room that is pre-requisite to the attainment of specification steam and without which he only multiplies and magnifies his problems.

In the utilizing system, in one method of operation, the pressure is essentially atmospheric at all times; in another, a vacuum is maintained alternating in value between a desired maximum and minimum. In a system operating at atmospheric pressure when the oxygen of the original air in the system has combined with the metal, leaving inert nitrogen, further replacement of the oxygen is dependent in considerable part on the gases from the steam. Thus, the oxygen available as a quantity factor in corrosion is relatively limited in amount if proper care is used in production of the steam.

In a system operating under vacuum, continuous removal of the gases in Division 3 is going on. In proportion to leaky connections, open traps, and other paths of entry for the surrounding atmosphere at higher pressure, air enters, and the oxygen concentration constitutes a considerable proportion of

the gases in the system. The data of Table 12 are conclusive on this point. The partial pressure of oxygen therein, and hence its tendency to dissolve in the condensate and thus become actively corrosive, are mitigated by maintenance of the vacuum.

In the former type of system, corrosion is more or less limited to the oxygen provided in the steam; in the latter, the relatively low partial pressure and hence restricted solubility of oxygen resultant from the vacuum is the limiting factor. In the former system, when steam is shut off over night, the non-condensable gases should be relatively poor in oxygen and should continue thus; in the latter, the non-condensable gases are well-nigh as rich in oxygen as the atmosphere, and hence if opportunity is given them to dissolve, they should be capable of many times the corrosive activity that would occur if they were largely oxygen-free at the start. In the former, tightness of all connections to the point of preventing gas inleakage is an excellent provision but is not an especial factor in the prevention of corrosion, since pressures within and without the system are the same; in the latter, each connection at which inleakage may occur becomes a potential source of supply for the quantity factor of corrosion. Therefore, in the vacuum system, since continuous maintenance of the maximum vacuum sustains the minimum partial pressure of oxygen in the system, tightness of all the multitudinous connections therein and prevention of oxygen entry through all traps or breather lines becomes a more necessary pre-requisite for prevention of corrosion than small differences of a few tenths ml/liter of oxygen in the entering steam.

Despite the amounts of oxygen that are available in the utilizing system because of its operating characteristics, corrosion therein to any troublesome extent is not general, and, as already noted, when it does occur, is usually confined to certain definite areas. This statement applies to both types of system which have just been discussed. Thus the return lines following the trap on the hot-water heater constitute one area of attack; the nipples connecting the radiator traps to the return lines constitute a point for collection of corrosion deposits, although, as illustrated in this investigation, these deposits had their origin probably in the radiator, whence they were transported to the point where found.

The total quantity of oxygen present in the system is not the deciding factor in the amount of corrosion that occurs; only that in dissolved form is effective. For instance, when the steam supply to a radiator is turned off and air is drawn in as condensation occurs, the clean walls of the radiator, drying quickly, are soon immune from attack, whatever the percentage of oxygen present. Over the limited area at the bottom, however, where the condensed water collects, corrosion proceeds. This is in keeping with the observations of Markson and Finnegan regarding a radiator removed from Building *A* for examination, in which a water-line at the bottom was plainly apparent, but corrosion over the surfaces was absent.

A second proviso, therefore, in protection from corrosion, is vested in such arrangement of the system that when the oxygen concentration is apt to be high, continuation of condensation on the surfaces shall not occur. An illustration of this fault was noted in the discussion of the nipple deposits from Building A. Also, drainage should occur as rapidly as possible.

Dry surfaces at the temperatures incident to the utilizing system do not

require protection from corrosion, even though the contacting gases contain considerable oxygen. When the surfaces are wetted, and oxygen is present, protective films over the surface of the metal constitute the main defense against all corrosion of all types.

Speller * notes "that steam return lines in systems using exhaust steam from reciprocating engines rarely show serious corrosion, apparently due to the protective effect of a film of heavy hydrocarbon compounds deposited on the inside of the pipe" and notes that "oiling the steam will afford substantial protection to steam piping."

Until recently, the alkalinity of boiler waters has not been exactly controlled; and also, carry-over of boiler water in the steam has been more frequently endured than corrected.

With these conditions obtaining, one is prompted to desire more comprehensive information, in order that decision can be reached whether, in the above instances, protection was afforded by the oil film, or by the maintenance of high *pH* value in the condensed steam by the carry-over of boiler water.

Tests on the advantages and feasibility of so employing oil are to be carried out by the New York Steam Corporation and by The Detroit Edison Company. The results will be awaited with interest.

It may be remarked that this use of oil is antagonistic to that cleanliness of surfaces requisite for rapid and complete drainage thereof. Whether an oil film can be maintained that is free from breaks, and that will protect when droplets of water remain on the surfaces of a radiator, for example, as the concentration of oxygen increases during off periods, must be decided by observation under actual operating conditions.

It remains to discuss how far protection from corrosion can be assured by careful choice of metal to be used in the utilizing system. For absolute assurance of protection, the nature of the material must be such that the service conditions it meets must constitute a minor factor in its service life.

In general, requisite attention paid to the quality of steam, to the tightness of the system, and to details in its arrangement such as that of preventing fortuitous condensation of hot vapors on cool surfaces contacting a vapor phase with maximum partial pressure of oxygen, renders unnecessary any general use of materials in the utilization division other than those now employed.

At points in this Division, however, that are particularly susceptible to corrosion, methods of moderating attack to a point of toleration by local use of corrosion-resistant metals seems possible and perhaps advisable. For instance, in the return lines immediately following the hot water heating tanks, the kitchen, the laundry or the mangle, corrosion usually takes the form of grooving or pitting. According to the experience of Markson, a small mechanical change was sufficient to correct conditions following the hot water heater in a typical case of corrosion at this point, and it may be that, in general, a mechanical solution of these cases will be satisfactory. On the other hand, substitution of copper or brass pipes for steel at such points may be advisable, with due attention, of course, to avoidance of bi-metallic contacts at the joints.

Because of the mechanism of formation of the deposits in nipples following

* Corrosion in Steam Heating Systems, by F. N. Speller (A. S. H. V. E. TRANSACTIONS, Vol. 34, 1928).

the radiator trap, namely, by transportation of the corrosion products from the radiator to the nipple—the utility of substituting brass or copper for steel nipples is not apparent unless the metal of the radiator itself be of non-corrosive characteristics. An answer to this condition consists in limiting the dissolution of metal at the point serving as origin of the corrosion product. Means therefor have been pointed out.

Acknowledgment

When the New York Steam Corporation initiated the investigation of the specific cases of corrosion that constitute the basis of this report, it was agreed that the report should be general, in so far as possible, in its application to corrosion in steam heating lines. With the wealth of accurately taken data provided by the Technical Division of the Corporation, the authors have had opportunity to study these problems on the basis thereof, and to apply thereto, particularly in the utilizing system, the laws of chemical equilibrium. The results have been highly educational, particularly those phases bearing on oxygen and carbon dioxide content in steam, their partial-pressure relationships therein and concomitant solubility in condensate derived therefrom, the different *pH* values that characterize condensed steam samples and condensate derived from the steam. It is our hope that this study will help to correct some of the fallacies that commonly arise from ascribing to the condensate formed from and in contact with the steam those properties found in condensed steam samples.

DISCUSSION

F. N. Speller[3] and E. L. Chappell[4] (Written): The authors have given a clear statement of the fundamental laws that govern the solution of gases in water and some of the relations these laws may have to corrosion in steam systems. In addition to recognition of the technical value of the paper, the authors should be complimented for this frank discussion of the problem and the presentation of the results of their extensive investigation of conditions in buildings supplied by a district heating plant in New York. Such an attitude tends to lead to a satisfactory solution of the problem.

Considering the problem in general: in any case of corrosion it is important to weigh all the factors involved. The resultant of all factors determines the rate, but this usually is controlled by one or two dominant factors. In steam condensate corrosion the main factors may be considered as:

(1) Dissolved oxygen content
(2) Dissolved carbon dioxide content
(3) Temperature
(4) Rate of flow of condensate
(5) Protective coating formed (from the condensate) in steam return system.

These factors, it will be noted, are associated with conditions external to the metal. The factors mentioned dominate perhaps in the order given, although their relative importance depends very much upon local conditions.

[3] Director, Dept. of Metallurgy and Research, National Tube Co., Pittsburgh, Pa.
[4] Department of Metallurgy and Research, National Tube Co., Pittsburgh, Pa.

It may be said that the following general facts have so far been established regarding steam corrosion:

(1) Corrosion occurs only where corrosive condensate is in contact with metal.

(2) Two distinct types have been identified, although they often occur together:

(a) Grooving on bottom of horizontal returns with clean metal surfaces—here carbon dioxide is probably the dominant factor. (In this type, corrosion products, if seen at all, are deposited away from the point of corrosion where they may accumulate to the point of clogging.)

(b) Rusting with or without pitting—here dissolved oxygen predominates. The pitting type occurs particularly where lines lay full of condensate, and where the rust is deposited at or near the seat of corrosion.

(3) Steam corrosion appears as an isolated and seemingly erratic phenomenon. This is true, both geographically and in regard to its occurrence in a single building. Corrosion may occur in any part of the country whenever operating conditions are such as to lead to the necessary combinations of factors. However, in districts having water high in bicarbonates, there is a decided stronger tendency to form steam high in carbon dioxide which is more corrosive. In buildings, corrosion ordinarily occurs only in certain sections of the horizontal returns, and is apt to be more severe where the pipe carries a large volume of condensate at high temperature.

(4) All ordinary ferrous metals, rolled or cast, have about the same corrosion rate, but there are data that indicate somewhat longer life for copper steel under acid conditions.

(5) The damage seems to be more acute at the present time in certain isolated sections than was the case 18 or 20 years ago. This does not seem to be due to a change in any single factor.

We agree with the authors' conclusions that severe corrosion in parts of the system referred to as operating below atmospheric pressure is at times due to inleakage of air. But we find it difficult to apply this explanation to parts regularly operating above atmospheric pressures, except for those cases where the pressure is frequently lowered for temporary periods. Where corrosion occurs in systems operating under pressure, we must therefore also consider the possibility that conditions (partial condensation or other causes) lead to much higher concentrations of carbon dioxide and oxygen at certain points in the system than those calculated by the authors, so that the concentration of carbon dioxide and oxygen, as they exist in the steam at times, may result in condensates which are corrosive of themselves.

It will be noted that the solubilities calculated are based on the assumption that the steam and condensate exist at equilibrium temperature throughout the systems, whereas under practical conditions it would seem that the condensate must be cooled below the prevailing steam temperature which will result in a considerable increase in the calculated dissolved gas concentration in the condensate. As much as 16 per cent carbon dioxide has been reported in dead ends in certain western systems using waters high in bicarbonates. It is evident that under practical conditions many intermediate conditions between dead ends and free flow may exist. We have seen reports of experiments showing corrosion under conditions of steady flow where inleakage could not occur.

The effect of carbon dioxide could well be emphasized to a greater extent. The authors discuss a case where it is present in the steam to some 27.5 times the weight of oxygen. In addition to considerations mentioned above, the fact that carbon dioxide is a gas which enters into chemical combination with water must be considered in connection with both the amount which will go into solution and the rapidity with which it will be released when conditions are favorable to this change. Solutions of carbon dioxide are true acids, partially ionized. and able to attack metals with evolution of hydrogen. The combined effect of acids and dissolved oxygen must be taken into account as well as the effect of acid solutions in preventing formation and causing destruction of protective films already deposited on the metal.

In interpreting the results emphasis should be placed on the actual concentrations of carbon dioxide and oxygen reported by the authors, as found in the condensate, which do not always agree with the calculated concentrations. It seems that a considerable increase in oxygen without a corresponding increase in carbon dioxide must indicate air inleakage, while a variation in content of carbon dioxide would ordinarily come from variation in the boiler water. It would be of value if the authors supplement the data given at certain points. For example, the amount of steam flowing is generally obtainable with fair accuracy, and in considering steam corrosion where the active constituents in the steam are quite small in amount, the total quantity of corrosive constituents becomes important so that we suggest the tables could well be supplemented with data as to pounds of steam per hour. Also, data as to the amount of corrosion noted, either as general observations or as specific corrosion rates under the various conditions investigated, combined with measurements of the oxygen and carbon dioxide in condensate, are needed before final conclusion can be drawn as to the permissible limits for these constituents.

The authors' comments as to the relatively small amount of piping affected should be emphasized. The larger portions of steam systems are not attacked, and, as discussed previously, local corrosion, as for example, at trap discharges, does not indicate that general failures in the steam system are to be expected.

More data are needed on the rate of attack of condensate under different conditions as found in service, including amount of oxygen and carbon dioxide in condensate and steam at points in system under observation. Such data and further experience will probably suggest the most economical remedy, but it is evident that a reasonable limit should be placed on the oxygen and carbon dioxide in steam; that vacuum systems should be maintained as tight as possible, perhaps by the more general use of welding; that more attention should be given to design of system with sufficient drainage and grade to all returns; and perhaps chemical treatment of steam to inhibit corrosion or cause formation of protective coatings on the metal.

It is well known that no corrosion is found in closed hot-water heating systems, so that under some circumstances a steam-heated hot-water system may be the solution.

RAYMOND NEWCOMB[5] AND W. R. FIDELIUS[6] (WRITTEN): This paper brings to light some interesting data on steam and condensate in the various phases of a heating system.

The authors have gone to considerable length in attempting to disprove the impression that the corrosion in certain heating systems is attributable to the steam as generated from a central plant where the condensation is not returned to the boiler. The paper is an admirable treatise on the Law of Henry and Dalton, and the solubility of oxygen and carbon dioxide.

However, the discussion and calculations to prove that the oxygen in the steam containing 0.5 ml per liter, which converted to cubic feet at 100 C amounts to 0.00001071 cu ft per pound, is interesting, but no one would attribute corrosion to this minutely small quantity. For instance, considering the radiator used in the discussion, containing 73.33 sq ft of radiation and a cubical content of 1.186 cu ft, it is seen that only one complete air change in 24 hr amounts to 0.237 cu ft of oxygen. Assuming the radiator to be condensing steam continuously at its rated capacity for a like period, namely, 24 hr, the total oxygen contained in all this steam of the quality indicated in this paper would amount to only 0.00470 cu ft. Considering moreover that the average operating conditions would result in filling the radiation with air several times in the 24-hr cycle, we believe that any oxygen which may be carried by the steam is a small part of the total air entering the heating system.

[5, 6] Fitzgibbons Boiler Company, Inc.

As stated by the authors, the rate of corrosion is a function of the *pH* value. From the data and analyses given, it is seen that the *pH* value is little or no function of the oxygen content in the steam or condensate. It is admitted, however, that the CO_2 content does lower the *pH* value and accelerate the corrosion. It is furthermore admitted that the corrosion is not done by the physical contact of the CO_2 which is carried over in the steam, but its effect is like that of a catalytic agent in that it accelerates the action and is not removed by the reaction.

Inasmuch, as the CO_2 content in the steam depends primarily upon the amount of feedwater treatment necessary at the generating station, and as stated previously by the authors that the lower CO_2 content will raise the *pH* value and inhibit corrosion, the justification of the authors in establishing the fixed value of the CO_2 as found in the samples taken as being sufficiently low, is questionable.

The element oxygen which does the actual corroding has always been available in heating systems and no doubt always will be. Heating plants also have not changed fundamentally and infiltration of air into this system is inevitable. Bearing this in mind, exception is taken to the paragraph made by the authors in their conclusions: *It would be egregious to suppose that avoidance or solution of these problems of corrosion resides in the mere substitution of an individual boiler plant in the building itself in place of district steam.* In a building containing its own heating plant the steam cycle is considerably different from that of a similar building employing district heating. It can be readily seen that the plant containing its own boilers is working in a closed cycle, the condensate returning directly back to the boilers. The CO_2 content in the individual boilers will be eliminated as there is little or no feedwater treatment necessary. Whatever treatment is made, the CO_2 formed by the treatment must, of necessity, due to the characteristics of the feedwater, be negligible considering the closed cycle in an individual plant, as there is practically no loss of the original water. This loss of the CO_2 content in the steam will, therefore, automatically raise the *pH* value and give a steam which will be less corrosive. The steam thus generated in the individual boilers is, as the authors state, not made to specifications, but it must be admitted that there is slight room for improvement in the quality of the steam generated from the distilled water which has returned through the condensate lines.

The data giving the CO_2 content and *pH* values of the condensate are evident proof that the CO_2 content of 19.6 ppm contained in the steam is practically eliminated through the vents in the low pressure phase of the system and, therefore, does not return to the boilers.

LEWIS JONES[7] (WRITTEN): We have established beyond question; *first:* That a water radiator supplied by the 1-pipe steam system and partly airlocked will stop up the valve and first nipple quickly with a deposit of iron oxide while under exactly the same condition a steam radiator will keep the valve and piping clear of obstruction; *second:* That if there is sufficient leak in the radiator to keep it hot all over it will not stop up the valve and nipple; *third:* A small radiator will not stop up the valve and nipple.

Our company was forced to go into this problem about 5 years ago and I think our results will prove of interest.

We heat about 1600 private dwellings. Most of them are equipped with 1-pipe steam systems and we strongly recommend the use of hand operated air valves on each radiator instead of automatic air valves. In mild weather these radiators can be kept partly full of air thus making the heat given off correspond to the amount required; thereby reducing the steam consumption and at the same time keeping the room at a more comfortable temperature.

[7] President, Lewis Jones, Inc., Philadelphia, Pa.

About 5 years ago we began to have trouble with the nipple next to the radiator stopping up with dirt which upon analysis proved to be chiefly an oxide of iron. This trouble increased until the heating systems in 500 houses were affected. A 1¼-in. nipple starting perfectly clean would nearly close in a week.

The trouble always occurred in systems installed in new houses. The heating systems in houses that had been built for several years worked perfectly although the piping design was identical with that in the new houses. These systems in the old houses were equipped with steam radiators (the loops not connected at the top) and the systems in the new houses were equipped with water radiation (the loops connected at the top). The heating systems in one block of houses, built along with the houses where this trouble was experienced, worked satisfactorily. Investigation disclosed that the houses where the radiators worked properly had steam radiation while the others had water radiation.

The trouble with nipple stoppage developed just after the radiator companies started to supply water radiation only and this has been found to be the cause of all this trouble.

For the last 3 years we have been taking many hundreds of radiators apart and re-assembling them with blind nipples across the top. Not one of these changed radiators has ever become stopped up.

We have found that a free-acting automatic air valve or a small leak in the radiator will remedy the stoppage trouble but we could not advocate this on account of the necessary excessive increase in our bill to customers.

Seldom did we experience trouble with small radiators.

Upon the discovery that this dirt that stopped up the nipples was almost entirely an oxide of iron we feared it indicated that our distribution system was being attacked by the steam. However, since we find this trouble can be eliminated by using steam radiation we think the iron came from the radiators exclusively.

So far, I have dealt with established facts. What follows is merely my hypothesis as to the cause and may be right or wrong.

A water radiator partly airlocked will heat entirely across the top while the bottoms of all the sections except one or two at the feed end will remain cold. This causes the condensate to trickle down the inside walls of the radiator and to be cooled nearly to the room temperature by the time it reaches the bottom. This low temperature permits the condensation to dissolve a large amount of oxygen and CO_2 while the moistened condition of the radiator wall at a constantly decreasing temperature is particularly favorable to attack from the condensation with its dissolved gases. However, as the condensation leaves the radiator through the pipe by which the steam enters, it is re-heated to the steam temperature and the gas is driven off and carried by the incoming steam into the radiator to repeat the process.

This driving off of the gases evidently causes the condensation to lose its ability to dissolve the iron at this point and it is precipitated and stops up the pipe.

Under the same conditions a steam radiator (loops not connected at the top) heats each successive section all over beginning at the feed end and the temperature of the condensation is never reduced appreciably below the boiling point. Consequently, it does not dissolve gas and does not attack the interior of the radiator.

As this objectionable deposit does not occur when there is an air leak sufficient to keep the radiator hot all over, the different effect might be due to the removal of the oxygen and CO_2 or to maintaining the condensation at a high temperature. I am inclined to the latter explanation because the air content of a water or a steam radiator when airlocked should be identical and the latter gives no trouble and because small radiators escape this trouble due to the transfer of heat by conduction. If one part of a small radiator heats, the entire radiation usually becomes warm.

We have not had this stoppage trouble develop in any 2-pipe system but believe it would occur if the cool condensation from one radiator should meet the overheated condensation from another radiator.

Radiators operated as I have suggested—partly airlocked—will gradually accumulate more gases received from the incoming steam but the proportion of each will remain about the same whether a small part or all of the radiator is airlocked. With this arrangement it is usually only necessary to let air out of the radiator when there is an extreme change to cooler weather.

H. D. NEWELL[8] (WRITTEN): The problem of corrosion in iron and steel pipe in steam condensate and hot-water heating systems has caused considerable concern and all too little attention has been paid to the influence of oxygen and other dissolved gases in the condensate.

With present central station practice feedwaters are kept below 0.05 ml oxygen per liter by suitable treatment and consequently it is believed that the majority of corrosion resulting from oxygen is due to inleakage in the piping system and not to steam quality. The authors' conception of oxygen as the capacity factor in corrosion and *pH* value (as affected by carbon dioxide content) as the velocity factor is a novel means of distinguishing the effect of these primary corrosive agents.

Mechanical strain introduced by bending or threading no doubt accelerates the normal corrosion rate by electrolytic action between strained and unstrained portions of the same pipe or fitting. Certain metals appear to be more susceptible than others in this respect and among the iron or steel products, toncan iron (open hearth iron containing copper and molybdenum) seems to be unique in that its corrosion rate is relatively unaffected by strain set up by hammering, flattening, bending, or threading.

Aside from the accelerated corrosion rate due to strain, there is a rapid acceleration in corrosion rate resulting from "mill scale". In black pipe or tubing, the corrosion at points such as straightner mark breaks and other broken points in the mill scale coating results in electrolytic action giving rise to localized corrosion which may be many times the normal corrosion rate of the base metal. Rapid pitting is the result and, therefore, consideration should be given the proper preparation of the surface of pipe before installation. In many cases pickled or scale-free pipe will minimize the corrosion and provide satisfactory service.

Concerning the chromium alloy steels, such as 18 per cent chromium and 18 per cent chromium 8 per cent nickel alloy, these are available in pipe or tubular form and in fittings, but their high first cost has somewhat retarded their use. These alloys depend on film formation for their corrosion-resistant properties and contrary to ordinary iron and steel are more corrosion-resistant in oxygenated water or under oxidizing conditions than in water free from oxygen or dissolved gases. The use of stainless or corrosion-resistant steel pipe might be profitable in certain locations that are particularly susceptible to corrosion.

The maintenance of a high *pH* value in condensate water as suggested in the authors' paper is one way of minimizing corrosion by reduction of the velocity effect.

S. E. TRAY[9] (WRITTEN): The spectre of corrosion, ever present where water is conducted through closed pipes, has always loomed large on the engineers' horizon as one of the numerous factors necessary to be considered in obtaining an ideal water supply. The application of the laws of physical chemistry and of chemical equilibrium to the prevention of corrosion in a manner capable of translation into normal operating practice is a definite step toward the solution of one of the most troublesome problems of steam generation.

---

[8] Chief Metallurgist, The Babcock and Wilcox Tube Co., Beaver Falls, Pa.
[9] Sargent & Lundy, Inc., Chicago, Ill.

Regardless of the ultimate use of steam, whether it be for heating, process work, or power generation, the problem of corrosion of pre-boiler equipment is one of the first order. If the make-up requirements are high the problem is intensified, but even with a large portion of condensate returns and small make-up the difficulty arising from various types of waters is sometimes overlooked. This is especially true of plants operating on Chicago city water or more generally, those numerous plants whose raw water supply is obtained from Lake Michigan.

LAKE MICHIGAN WATER PPM

Bicarbonate (HCO_3)	136.6
Sulphate (SO_4)	18.0
Chloride (Cl)	4.4
Silica (SiO_2)	4.5
Iron (Fe)	2.0
Calcium (Ca)	35.2
Magnesium (Mg)	10.2
Sodium (Na)	9.6
Total solids	176.0
Available CO_2	98.5

Compared with Croton, Catskill, or Detroit water, chance for corrosion with Lake Michigan water is greatly amplified. An average analysis, given in the accompanying tabulation, shows the bicarbonate or available carbon dioxide to be twice that of the average North American surface waters and over 40 per cent greater than that found in Detroit city water. Complete treatment with phosphate is usually uneconomical since the theoretical requirements are 142 lb of trisodium phosphate per million pounds of water. Partial treatment with lime and soda ash followed by phosphate to eliminate the last traces of residual calcium appears to be a more logical consideration.

A number of central stations and some industrial plants in the Chicago district make use of evaporators to supply distilled make-up to the condensate lines. They are installed without pre-treatment, other than filtration, and in many instances the vapor supplies additional heat to the boiler feedwater before it is condensed. The half bound carbon dioxide liberated with the vapor is dissolved again when the vapor condenses, with the result that the pH value of the evaporator distillate is of the order of 4.5, an actively acid water. Analyses show the content of free CO_2 to average 45 ppm and the Law of Henry and Dalton indicates the cause of the acid water to be due to the high partial pressure of the carbon dioxide.

Although the amount of make-up is usually small when evaporators are used, the continual introduction of such acid water is sufficient to keep the pH of the boiler feedwater well below the value of 9.6. This, in turn, means continual evolution of hydrogen and dissolution of the metal even in the absence of oxygen, a fact which the authors have well established.

The remedy for this type of corrosion would appear to be pre-treatment of the evaporator feed by lime and soda ash to eliminate the source of the carbon dioxide. The addition of caustic soda to raise the pH value of the raw lake water is not entirely successful due to the buffer action of certain soluble salts found in the water. When certain conditions make pre-treatment inadvisable the way out appears to be by the introduction of caustic soda to the boiler feedwater in the manner which Mumford and Markson have developed at Kip's Bay.

In the application of the Law of Henry and Dalton, the influence of the nature of the gas on its solubility should not be overlooked. In general it may be said that those gases which exhibit acid or basic reactions are the most soluble, the solubilities of neutral gases being small. While this may not be a factor in the behavior of natural waters, certain exceptions to the law may well be pointed out. For plants

operating on river waters high in organic material the presence of ammonia is of serious consequence, resulting in corrosion of brass valves and fittings of radiators with resulting deposition of ammonium carbonate. Ammonia gas is extremely soluble and has marked basic properties, but at ordinary temperatures and up to 212 F does not obey Henry's Law, as the mass of ammonia absorbed is not proportional to the pressure. The resulting high concentrations of ammonia found in some condensate lines may therefore be the result of absorption of the gas at the feedwater heater if the partial pressure of the ammonia is not materially reduced.

E. B. RICKETTS[10] (WRITTEN): In the face of the evidence presented, it is difficult to see how the authors' conclusions can be successfully questioned. The authors have shown by conclusive analytical data that the oxygen and carbon dioxide content of the boiler feedwater were reduced to harmless proportions before being fed into the boiler and that the chemical treatment added was of such a nature that any entrainment would tend to reduce rather than accelerate corrosion. It is obvious that if the feedwater enters the boiler free from corrosive agents there is no way in which they can enter the steam until its pressure is reduced to below that of the atmosphere, which can never take place until it enters the heating system at the point of use.

It is theoretically true that a boiler installed in a building, where the heat is used, could operate on a large percentage of condensate for boiler feed, and this condensate, if properly protected from outside contamination, should be practically free from corrosive agents. The probabilities are, however, that due to the absence of skilled chemical supervision of the feedwater, the steam produced in such a plant would carry a much larger content of corrosive gases than would be found in steam from a district steam station where the quality of the water is watched night and day. Oxygen enters a low-pressure system in all sorts of unexpected places and can only be kept out of the boiler feed by constant vigilance on the part of skilled chemists.

Those of us who have operated power plants know that the oxygen entrained in the steam never presents a serious problem while the fluid is under a positive pressure, however, as soon as either vapor or condensate is under a vacuum our troubles begin. It is much easier to maintain a system tight with 1000 lb pressure, than with 5 lb vacuum and the only sure way to eliminate gases from the boiler is to take them out just before they enter the boiler feed pump.

A radiator system which is alternately full of air and steam is obviously an ideal apparatus for absorbing oxygen and from the data presented by the authors it works just the way it would be expected to work.

R. M. GATES[11] (WRITTEN): In reviewing this paper, particular stress should be placed on the thoroughness with which the information presented has been obtained. Dr. Hall's original report from which this paper was prepared showed that his effort has been most exhaustive and, as far as our experience goes, quite novel. Some of the ideas presented can be followed with benefit in determining problems as encountered in other engineering fields.

In detailed discussion of this paper, particular care must be taken in conditions stated. If conditions are at variance with those under which the results given have been obtained, these should be noted in an explanation.

N. ARTSAY[12] (WRITTEN): The authors have given the clearest explanation of the corrosion problem and their searching analysis of all the numerous factors entering in it. There are many queer theories current mostly due to the lack of proper information. For instance, in Germany there is an idea that in high pressure boilers

[10] Research Engineer, New York Edison Co., New York, N. Y.
[11] Superheater Co., New York, N. Y.
[12] Geo. A. Orrok, Consulting Engineer, New York, N. Y.

the saturated water decomposes and there is a "hydrogen corrosion", facts notwithstanding.

The authors made a good starting point by dividing their problem into 3 major parts: the properties of steam, the properties of condensate and the service conditions. The effect of Dalton's Law on oxygen content in condensate is well taken, however it is interesting to get more theoretical data on variations in oxygen content in the case of intermittent condensate discharge and a degree of undercooling of condensate in respect to saturation temperature. The paper also does not give the order of pH value changes with temperature. The values given are for room temperature. It would be interesting to know the order of variations up to 180-200 F.

The harmless nature of commercial steam and its condensate in normal operating conditions is clearly defined both from the velocity and the quantity factors of corrosion. Experience with feedwater heaters in the central station practice substantiates this deduction. While the oxygen content in power station steam is usually low 0.05 or less, the feedwater heaters receiving steam bled from low-pressure stages and glands are dealing with substantial amounts of air and CO_2 but no corrosion of any serious extent has been observed. Most of the heaters have vents for non-condensible gases and the amount of CO_2 in one case was so high as to cause suction troubles in booster pumps serving direct contact heaters.

The influence of variable service conditions is set in the paper very mildly. My 15 years' experience in the former Russian Navy, the last 5 years of which concerned maintenance and repair of ships ravaged by war and revolution, has taught me the utmost importance of steady conditions of environment for a long life of steel. During the Civil War in Crimea, a battleship laid up for 11 years had to be commissioned in 4 days, her scotch marine boilers 28 years old were found in excellent state due to proper drying, when laying up, and absence of changes in inner atmosphere. On the other hand, we arranged a crew of divers which rescued from the mud bottom of the bay of Sebastopol 1200 rifles which were under water 3 years. If cleaned and oiled within 15 min after being brought to the surface, the rifles were serviceable and were sent immediately to the battlefront. A delay of 1 hr in drying them produced such pitting that a thorough overhauling in the armory was necessary. The same crew of divers found a British cutlass of damask steel lost in the bay during the Crimean War in 1855. The blade was in excellent shape and even the polish was not affected. The change in surrounding conditions is the greatest accelerating factor of corrosion. Alternate humidity and dryness or wetness is probably the best accelerator of corrosion in the presence of oxygen. The authors neglected the dry corrosion (unaccompanied by dissolution) in humid atmosphere present at times in radiators and piping, while my experience with ship hulls indicates its serious extent, especially in presence of Fe_3O_4 scale.

There is no question in my mind about the major factors of corrosion in systems involving condensing steam and steel surfaces, the CO_2 and O_2 in steam and condensate are distributed so evenly in the fluids that any localized corrosion is out of question, while the general dissolution of iron will be felt only after several decades. It is the local changes of environment in a variety of forms, thermal, mechanical, electrolytical, etc., which produce the local destructions by corrosion, and, regarding a system like the central heating system, I would expect corrosion in places of intermittent action. Let us consider, for instance, the condensate discharge lines. There will be at least 30 F changes at trap discharges involving thermal stresses of considerable amount, up to yield point in favorable spots accompanied by leakage of atmospheric oxygen, and we know that variable stresses accelerate corrosion. If I had a vacuum heating system and were forced to operate a vacuum pump to handle air leakage, I would expect corrosion. For an evaporator blow-down line on a battleship, I have put a welded steel pipe 1½ in. inside diameter with walls, 1¾ in.

thick, which lasted 3 years, and the first consideration for the unusual thickness was the violently intermittent action of the blow-down with the attendant disturbances in stress and temperature.

The present steam heating systems in buildings are designed without any regard to difference in the rate of corrosion in their various sections and a heavy penalty is paid for this omission. Thus some parts of the system are becoming troublesome long before the system as a whole suffers the depletion of margin in the thickness of steel.

W. A. SHOUDY[13] (WRITTEN): Some 30 years ago the majority of steam generating stations were equipped with closed feedwater heaters. At that time it was necessary to use brass piping for feed lines because of the rapid corrosion of iron or steel piping. When open heaters were installed they were all equipped with a vent, principally to avoid the collection of air at the top of the heater, thereby reducing the feedwater temperature. This air came from the feedwater.

Because of the increase in the size of feed piping, brass piping became too expensive and we returned to steel and cast-iron pipe. In many cases the corrosion of this pipe was negligible but in dead ends at times corrosion was rapid. I have seen such piping that has been corroded in 3 months to look almost like a colander. So long as the usual velocity of the water was in the neighborhood of 3 ft per second, the corrosion did not occur.

When closed heaters were used, there was no opportunity to relieve the water of oxygen and as this water was heated the high percentage of oxygen immediately attacked the feed line.

With the open heater, a large part of the oxygen was boiled off and the remainder could not free itself readily and could attack the metal only where the velocity was low, or where there was a sudden drop in pressure as in the entrance to tubes or in sudden changes of direction. Where there was no dead end in the piping, the oxygen was carried into the boiler and set free generally near the water line and we had evidences of pitting.

With the introduction of the large tube steel economizer, serious corrosion occurred immediately because the oxygen was set free by heat and because the velocity of the water in the economizer was not high enough to prevent the bubbles adhering to the metal. This condition was met by a closer study of oxygen and feedwater and to prevent further corrosion deaerators were installed.

The elimination of corrosion in the economizer and boiler did not end the trouble from oxygen. The small percentage of oxygen in the feedwater did not have an opportunity to attack these surfaces. It was carried over in the steam and on reaching the moisture zone of the steam turbine attacked the blades.

With the most perfect deaeration, the oxygen content often increased. Investigation finally showed considerable air leakage in the condensate pump shaft stuffing boxes. Re-design of these stuffing boxes has made possible the satisfactory control of the oxygen content, and corrosion is no longer a serious trouble.

J. W. DEGEN[14] (WRITTEN): This paper undoubtedly is of scientific value and it is interesting to make the observation that the primary reason for the preparation of the paper was the trouble being experienced by New York City buildings using street steam—the trouble being not only in heating return lines but also in the return lines from hot water heaters.

The paper seeks to establish through a means of abstract reasoning, involving mainly the application of the Law of Henry and Dalton, the specific limiting maximum

[13] Consulting Engineer, New York, N. Y.
[14] Supervising Engineer, Equitable Office Bldg. Corp.

amounts of oxygen and carbon dioxide that may be contained in the steam without rendering the condensate formed therefrom corrosive in an active measure.

This is in line with and amplification of Doctor Hall's more general statements in previous papers [15] on corrosion. In that report it will be noted that non-condensible gases in the steam are acknowledged as of possible damage promise but the specific allowable quantities are not estimated or guessed at. However, in a paper presented before the *American Society of Mechanical Engineers* [16] under the title Present Tendency of Boiler Water Conditioning, Doctor Hall does set some definite criteria and advance some more clearly delineated recommendations.

These statements are mainly interesting because of their general specifications on the responsibility of the conditioning process and also in the specific *pH* value of 9.6 given as a requirement to inhibit corrosion.

The present paper goes into considerable detail and analysis in an effort to prove that the quality of street steam as furnished by the central station in New York City is not such as to be the cause of corrosion of condensate return lines. By the fact that any study on the subject was made, there is admitted that such corrosion does exist to a serious extent.

Its existence is of interest to the designing engineer, the central station and the manager and owner of the building using the steam. in the inverse order of their mentioning. The designing engineer wants to correct in the design those faults in it that can be blamed for any part of the trouble. The central station wants to maintain a clean reputation mainly as a necessary measure in keeping satisfied customers on their service. The building manager and owner want the trouble remedied regardless of blame. The building bears the brunt of the difficulties. It is faced, in the worst extreme, with the job of the entire renewal of the return-line system far short of its expected life. At best, the job is one of incessant renewal of failed pieces of pipe, with the restoration of that part of the structure damaged by the condensate leaking out. It is a continual round of small repair jobs on the pipe system; replastering jobs on walls and ceilings; painting jobs on the plaster patches —it never ends. More than anybody he appreciates the seriousness of the problem not the least part of which is placating inconvenienced tenants. For all these three, the matter of laying the ghost of the actual cause or causes in their proper share is of interest.

As a matter of practical consideration, the building manager or owner who is faced with corrosion troubles in the return lines of his heating system or kitchen steam system must plan his course to eliminate the trouble—regardless of what the cause. Mainly because he is too busy with his other duties he cannot be depended upon to properly apply the Law of Henry and Dalton. He might try applying it to the outside of the pipes with a brush. He may with some degree of sense decide that it is like the Prohibition Amendment and should be ignored.

As a result he looks for a definite course of action—a definite solution—not a course of study. He immediately reaches the conclusion that—

1. If the quality of the steam is at fault, he will insist on correction, but if this is unattainable he must look to a remedy.
2. He may look for an ally to mix with the steam to inhibit the corrosion.
3. He may try a different pipe material provided the cost is not too great.
4. If he takes the trouble to make inquiries of his neighbors he might conceivably find that mainly and in general the only others in his same predicament are those who use street steam. He will find that his neighbors who have isolated power plants and those with boiler heating plants are almost untroubled by this particular difficulty.

He then can take his choice as to procedure—maybe adoption No. 2—the inhibiting agent if he can find a satisfactory one until such time as he can more fully study

[15] See Boiler Water Conditioning with Special Reference to High Operating Pressure and Corrosion,—(Midwest Power Conference 1926).

[16] See *A. S. M. E. Transactions* (Fuels and Steam Power Section 1929).

the possibilities of the No. 4—his own heating plant. It may be that he can bring sufficient light to bear on this latter to give a true picture, in which case he will find that, the published propaganda to the contrary notwithstanding, the troubles of the individual plant are as naught compared to those entailed by street steam service where corrosion conditions are rampant. The result will be the installation of an individual plant, if the study of costs shows any results within reason.

In the paper there are statements made and figures given on which some further enlightenment is to be desired. It is stated that though the dissolved carbon dioxide sets up in the water a low *pH* value that is accelerative to dissolution of the metal there is no sustained dissolution because the dissolution in itself supplies the ferrous iron to increase the *pH* value. This is true only where the water under question does not change. In the operation of the system however it is constantly being replenished thus sustaining a low *pH* value with consequent sustained dissolution.

Various samples of condensate were analyzed for their *pH* value. These values ranging from 4.7 to 6.7 as listed in the tables average 5.3. Doctor Hall has said—"even with the condition of perfectly deaerated distilled water, the *pH* value should not fall below a minimum of approximately 9.6 to avoid slow dissolution of the metal if not pitting". In the light of this and of the analyses and computations that show an actual content of oxygen and carbon dioxide to some degree in the condensate and the relatively low *pH* values how is Doctor Hall's previous recommendation to be reconciled with the present paper which states that these low values are not of primary significance? The previous statement has the advantage of concurrence by other authorities. It would seem then the better course would be not to minimize the low *pH* values but to recommend their increase by some means. Boiler carryover would do it and has done it but is objectionable from many standpoints.

Table 3 gives actual test data on feedwater temperatures and oxygen content. For what time percentages do the feedwater heaters operate in the lower temperature range with the corresponding higher oxygen content—which in some cases is given as high as 1.3 and 1.7 cc per liter. The paper is vague on this point. No definite statement is made that the feedwater heater temperature is uniformly maintained above 210 F or that the oxygen content is kept below 0.3 cc per liter although the maintenance of these conditions is inferred from the content.

In the discussion of the solution of the gases under the application of the Law of Henry and Dalton, no quantitative value is given to the effect of the area of exposed liquid with its accelerating effect. The area may be tremendously increased by turbulence such as the discharge of a bucket trap where as the condensate and oxygen are blown out they are intimately and thoroughly mixed.

In the conclusions the Law of Henry and Dalton is being decidedly overworked when it is used as an indirect reason to try to prove or at least intimate that corrosion is no greater in a building using an individual heating plant than in a building using street steam. The proof of this condition can be found only in an actual physical study. Thus far the most vociferous complaints on corrosion have come *not* from the individual plant building but from the newer of the street steam buildings. A more thorough investigation should be made before stating "that it would be egregious to suppose that avoidance or solution of these problems of corrosion resides in mere substitution of an individual boiler plant in the building itself in place of the distinct heating steam". To quote a word that appears frequently in the paper the authors' conclusion is "absurd". It savors more of commercial propaganda than engineering study.

As to the question of the consumer adding to his problems the "obtaining of skilled and understanding operation in his boiler room": Judgment of the correct-

ness and feasibility of the adoption of an individual plant rests not on the question of the troubles stated to be encountered but rather in the question of applied economics. Troubles are what the building agent or manager is paid to take care of.

In addition to the reference made to the use of steam oil atomizers to minimize corrosion and the New York Steam Corporation and Detroit Edison Co. experiments on its feasibility and advantages, it is enlightening to note the following references from convention *Proceedings* of the *National Building Owners and Managers Association* covering the past five years.

In the proceedings of the 1926 convention there were reported by Mr. Swetland of Cleveland the cases of two buildings that for years had been heated by exhaust steam from their own power plants with absolutely no corrosion troubles. A change was made to central station steam, and shortly thereafter, the failure of the return lines by corrosion started. The installation of an oil atomizer in the main steam supply remedied the condition.

The 1927 proceedings of the same association contain two references to the same subject. Mr. Folsom of Seattle reported the troubles of buildings that had purchased exhaust steam from a plant operating reciprocating engines, which were later replaced by turbines. When the reciprocating engines were used no troubles from corrosion were experienced. After the change to turbines, the failure of return lines began on a wide scale. Oil atomizers were installed and again proved an effective remedy. Mr. Malm of Cleveland in the report by the Research Committee on their study of corrosion suggests as a practical method for retarding corrosion the application of oil to the system preferably by means of oil drips to the tops of risers.

The 1928 proceedings contain under the Operating Methods and Devices Committee report the information that "in a number of places where the buildings have used central station steam, the managers have eliminated much pipe corrosion by putting in oil droppers at the tops of the risers".

In the 1931 proceedings Mr. Malm of Cleveland in a paper entitled A Chemical View of Office Building Operation goes into, at some length, the question of corrosion of steam and return lines. In part he advocates the removal of dissolved carbon dioxide and oxygen in the feedwater by deaeration or open feedwater heaters; the maintenance of a tight system to prevent air infiltration; the removal of the carbon dioxide in partial chemical fixation by calcium hydroxide treatment with subsequent filtration; the discontinuance of water-treating chemicals that release carbon dioxide at boiler pressure and finally the introduction of refined paraffin oil in steam lines where the condensate is of course not returned to the boilers.

In particular, Mr. Malm makes the observation: "Steam service from district heating plants has, to a great extent, eliminated the isolated plant and the prevention of corrosion becomes a problem for the public utility. Corrosion, which arises from air leakage in the heating systems of office buildings, is subject to treatment by the individual property owner, but the damage which is sustained as a result of acid steam cannot be easily prevented by the building owner as the carbon dioxide originates in the generation plant of the public utility from the use of raw water or boiler compounds".

These instances are cited as an enlightenment by actual past results on that portion of the report presented for discussion which refers to the experiments being carried on by the New York Steam Co.

The observation can be well made that the oil treatment, while undoubtedly efficacious, is not what can be called for want of a better definition "scientific". It can easily entail troubles from stoppages in lines and from so filling sylphon bellows as to render them inoperative. It would seem that the introduction of a soluble caustic to raise the *pH* value to higher than 9.6 would be much preferred.

The general conclusion formed by the three most interested parties as previously mentioned must be along the lines that indicate the part in which each particular phase is to blame.

The designing engineer can take to heart the question of the proper connection of the returns from the high and low pressure systems and the continuous and intermittently operated portions of the utilizing system, in the light of its operation.

The central station can continue to apply the Law of Henry and Dalton if it so desires, but the more logical step would seem to be to supply steam of a quality (not necessarily purity) to inhibit all possibility of corrosion troubles being laid at its door.

The building owner and manager can take care to have his system as tight as his next-door neighbor who operates an individual plant. He can do his own experimenting on an inhibiting agent to introduce into the steam he has bought. He can, if his studies so indicate, decide to abandon street steam and operate his own plant. He must get results—he cannot temporize—he alone sits in intimate contact with the trouble.

But all may well hold the reservation in mind that study of actual test pieces of steel and iron subjected to the corrosion effect of condensate of varying carbon dioxide and oxygen contents and at varying pressures, both above and below atmospheric pressure, and with varying pressures on the surface of the liquid will give more concrete results than abstract study even though based on primary principles. The main reason is that in the paper no definite corrosion rates for the varying solutions are cited from actual tests.

F. R. OWENS[17] AND H. E. EINERT[17] (WRITTEN): It is apparent that this paper has involved a considerable amount of work and the discussion in itself, we believe, is comprehensive and the references to the fundamental laws of chemistry are exact up to certain limitations; we believe that the applicability of the Law of Henry and Dalton beyond a certain temperature is argumentative. However, we cannot help but feel that the scope of the work has been restricted. By this we refer to the fact that the authors have carried their discussion to the utilization of the steam generated in the central heating plant by the utilizing equipment only. The discussion might easily have involved not only the utilizing equipment of various customers, but also, condensate return lines that are invariably involved.

We gather that 20 ppm of carbon dioxide and 0.3 cc of oxygen per liter may be considered normal limits for feedwater and condensates. Apparently, the authors have considered these limits as final, based upon the ease with which the operators of the heating plants might expel these gases and based upon their solubilities as may be calculated by the Law of Henry and Dalton.

Case I. Under the question of carbon dioxide, we have had the opportunity of being in contact with an underground supply which was subsequently used in condensers used for certain plant processes and which water proved to be very corrosive. A series of 5 runs covering a period of 2 months, in which the oxygen and carbon dioxide were determined each 15 min for 8-hr periods, proved that in each instance the oxygen content, as measured by the Winkler test, was zero and the CO_2 content ranged from 29.4 ppm to 33.6 ppm, with an average content of 30.7 ppm, as determined by the *A.P.H.A.* Method. The temperature of the inlet side of the condenser was 72 F and the temperature of the outlet side was 76 F, at 45 lb gage. Exceedingly severe corrosion was encounterd, in fact, it was that severe that cast-iron condenser heads had to be renewed on the average of every 3 to 4 weeks.

Corrosion difficulties were not only experienced in the condensing equipment, but also, in the suction and discharge lines of the pump. Initially the suction line to the pump was a ferrous alloy and the pump casing itself was of cast-iron. These

[17] Cyrus Wm. Rice and Co., Inc., Pittsburgh, Pa.

lines and the pump casing were severely pitted after a relatively short period of operation. Finally, the lines from the well were replaced with brass and even this alloy suffered severe corrosion after a period of about 2 months' operation.

We are quoting the following analysis, which is an average of the several analyses of this particular well water:

	Grains per U.S. Gallon		Grains per U.S. Gallon
Temporary hardness ($CaCO_3$)	1.44	Magnesium sulphate ($MgSO_4$)	nil
Free Soda	0.12	Sodium sulphate (Na_2SO_4)	1.08
Total dissolved solids	5.05	Sodium chloride (NaCl)	1.10
Suspended matter	0.47	Iron and alumina (R_2O_3)	0.23
Calcium bicarbonate ($CaCO_3$)	1.24	Silica (SiO_2)	1.10
Magnesium bicarbonate ($MgCO_3$)	0.17	Permanganate consumed (organic matter)	nil
Sodium bicarbonate ($NaHCO_3$)	0.20		
Calcium sulphate ($CaSO_4$)	nil		

The inference to be drawn from the foregoing problem is that even with the complete absence of any oxygen and with the presence of an appreciable amount of carbon dioxide, severe corrosion may follow. We are of the opinion that the foregoing citation is unique, inasmuch as it has been the first water that we have encountered in which it was definitely ascertained that oxygen was entirely out of the picture.

Case II. At another plant we initially found the oxygen content in the feedwater supply to average 0.15 cc per liter, with the carbon dioxide 2.0 to 3.0 ppm. The *pH* value of the supply averaged 7.0, as determined with Hellige-Klett apparatus and indicators. These figures are the result of tests taken daily over periods of many months. Corrosion in the boilers at the plant in question was severe and extreme penetration was invariably encountered both in the tubes and in the drums. From a chemical characteristic, the quality of the boiler water concentrates was extremely favorable as a phosphate plus a caustic soda treatment was being applied. The *pH* value of the boiler water concentrates invariably ranged between 11.0 and 11.5, indicating the presence of more than sufficient hydroxyl ions, such that the activity of dissolved corrosive gases would be depressed to a minimum, from a chemical protective viewpoint.

After determining the sources of leakage of oxygen or infiltration at various points in the feedwater loop and correcting the equipment involved for such leakage, the oxygen content then dropped to a maximum of 0.04 cc per liter and an average of 0.02 cc per liter, with the *pH* value and the CO_2 remaining approximately the same. Since the infiltration of air into the feedwater system has been stopped, test specimens in the generating equipment have shown a decided improvement, and also, the evaporating surfaces themselves now indicate practically no active corrosion.

Case III. We are in contact with a plant operating at 400 lb using zeolite make-up in which part of the steam is bled from high-pressure turbines and used as a heating load for a group of large and small buildings. A carefully controlled *pH* value of the feedwater is being maintained by the recirculation of a certain amount of boiler blow-down water in order to depress any corrosive activities within the feedwater loop. The oxygen content of the feedwater, most of the time, is a trace, but occasionally reaches as high as 0.05 cc per liter, while the *pH* of this same water is maintained at not less than 8.4. The average of the daily analyses of the steam that is used for the heating load, determined over a period of about 4 to 5 months, shows that the *pH* value of this steam ranges from 6.9 to 7.2 with a trace of CO_2. The oxygen content checked that of the feedwater. The *pH* value in this particular instance was determined by the LaMotte comparator and indicators and the oxygen and the CO_2 were determined by the methods as referred to above. The analysis of the condensate from this particular source showed a *pH* value of 6.5 to 7.0; CO_2

a trace; the oxygen ranged from a trace at certain places to as high as 4.0 cc per liter in others. The oxygen content was directly traced to air-operated Templeton traps used to inject the condensate in the main condensate line. Test pieces that were installed in the horizontal condensate lines of both air-operated and steam-operated Templeton traps showed that after a service period of 74 days, that no penetration whatsoever was found in the test pieces placed in the condensate line, indicative of the steam-operated traps, while in the case of the condensate lines from the air-operated traps, the maximum penetration of test rod was 0.033 in. per year. This indicates that it is of paramount importance to have as low a free CO_2 and oxygen content in the steam as is possible. With the oxygen content as previously cited, the maximum penetration of 0.033 in. per year may be considered damaging. The pressure of these lines was atmospheric and the temperature varied somewhere between 120 and 160 F.

From the paper we gather that the authors favor the ordinary type open heater. Apparently they believe that the deaerating type of heater represents an unnecessary expense in installation and operation for heating plants, while the ordinary closed type of heater would permit the entrance of too large amounts of oxygen. If the same reasoning as applied by the authors were applied to *Case II*, cited, where the maximum oxygen content was 0.15 cc per liter, based on the Law of Henry and Dalton and with a consideration of the pressure and temperature, the solubility of the oxygen within the boiler blow-down water, in our opinion, would have been less than that cited in Table 10. Careful inspections revealed that corrosion did take place.

If a limit of 1.0 cc of oxygen per liter will give a solubility of 0.000153 cc per liter at 165 lb, 366 F, then an original oxygen content of 10 cc will only give 0.00153 cc per liter. Whether the oxygen content is 0.000153 cc per liter or 0.00153 cc per liter, makes little difference. Based upon the reasoning in this paper, it would lead us to believe that the question of deaeration, as far as oxygen is concerned, is of little importance and it would be unnecessary to proceed to limits of 0.025 and 0.05 cc per liter of oxygen in the feedwater. Practical experience by those directly connected with power plants certainly has proved otherwise.

On this basis, therefore, it appears as if the Law of Henry and Dalton may not apply at higher temperatures and pressures and, furthermore, that the extent of the rate factor as outlined by the authors is not sufficient. In other words, it appears to us that the rate factor not only involves hydrogen ion concentrations, but other influences such as temperatures, the rate of diffusing of oxygen to the cathodic areas, stress, low and high cycle frequency, quality of the metal, protective films as related to oxide films originally present on the material (pipe) and stray currents. It is apparent that the paper under discussion has overlooked the last 6 influences mentioned and in our opinion all are of major importance with respect to the net damage which may result to the utilizing equipment and the return lines that are involved in steam heating practice. In this connection, we refer to the work of Evans and McAdams for more complete information. As previously indicated, the authors have not considered condensate return lines. Naturally, it is altogether possible that at certain points in the return system, the concentration of dissolved gases in the condensate will approximate that at the point of deaeration in the feedwater loop.

This leads to the question as to what we believe should be the limits of non-condensible gases. The amount of oxygen that may be present in the feedwater is, of course, at all times a function of the particular station under question. In the case of heating plants, inasmuch as corrosion in condensate lines is a major problem, we believe that the oxygen content should not be greater than that which can be removed with the best and latest type of deaerating heaters. Reputable manufacturers now guarantee that the oxygen content will not be greater than 0.025 cc oxygen

per liter and state that they can maintain these conditions economically. This should be the maximum limit.

As to the question of free CO_2 in the feedwater, the complete removal of this is a function of the deaeration, and hence it is our belief that the CO_2 content of the feedwater should be zero and certainly not greater than a trace. As to the free CO_2 content of the condensed steam or condensate, this is not a function of the deaeration, but rather a function of the quality of the water. If the alkalinity of the raw or make-up water is that high that an appreciable amount of CO_2 will be encountered in the condensed steam, then proper conditioning internally or externally of the water should be resorted to in order to insure that this CO_2 will be within the lowest possible limits. Any treatment, mechanical or chemical, which will approximate zero oxygen in the feedwater and zero carbon dioxide and zero oxygen in the condensed steam reaches the ultimate goal.

The quality of the condensate itself after its utilization, involves the equipment through which it has passed and is passing. The amount of dissolved gases, as indicated by the authors, naturally can be increased if the equipment involved permits infiltration of air. If limiting factors are resorted to in the treatment of the feedwater supply with respect to deaeration, and also, limitations are placed on the quality of the steam with respect to its chemical conditioning, such that the dissolved CO_2 and oxygen are maintained at a minimum, naturally, it is absolutely essential that all leakage in the customer's utilizing equipment and the return lines involved must be maintained at a minimum.

W. W. TIMMIS: The matter of corrosion has been of peculiar interest to me because of its effect on the operation of radiator traps. When corrosion forms in a heating system, it tends to lodge in and be deposited on the interior of the radiator traps. The cone valve trap will, to a large extent, crush, push aside and pass through such lodgement and deposit, at least to the extent that the steam leakage through the traps, caused by the presence of this corrosion deposit, will not be so excessive as to make it impossible to operate the system satisfactorily. With the flat disc valve, however, this corrosion deposit in heating systems, where it occurs to an excessive degree and where there is a marked tendency for the deposit to occur in the trap bodies, will prevent proper seating of the traps to such an extent that the resulting steam leakage into the returns will make it impossible to operate the system economically.

When this occurs we are called upon to explain why the traps are leaking. If we suggest that the traps be cleaned, it is generally made plain that we undertake the cleaning. In some cases we have done it if only to prove that the traps themselves were functioning properly. In several cases we have had analyses of these deposits made by independent testing laboratories and in practically every case the report was, "high percentage of ferrous oxide, indicating presence of excessive free oxygen". We have also made an intensive study of the matter ourselves, both in the field and in our own laboratory.

The building owner, through his engineer, has three alternatives: *First,* to clean the flat disc traps at intervals frequent enough to keep them free of deposit; *second,* to use a type of valve in the trap which will work reasonably well in spite of the corrosion or; *third,* to prevent the corrosion.

Some two years before the presentation of the paper under discussion, we came to the conclusion that free oxygen is largely responsible for corrosion in heating systems. I personally have seen some rather striking demonstrations tending to prove the truth of this conclusion. In several buildings where excessive corrosion was being experienced, this condition was practically done away with by making the piping system tight. There have been a number of instances similar to the following:

Two apartment houses of almost identical size were built by the same owner; one at 150 East 48th St., and the other at 2 Beekman Place, New York City. Both get steam from the same source, namely the Kips Bay Station of the New York Steam Corp. They are within a stone's throw of each other. The same engineer designed both systems and the same contractor installed them both. Corrosion at 2 Beekman Place is excessive and very troublesome, while at 150 East 48th St., there is no evidence whatsoever of corrosion. The system at 150 East 48th St. is tight enough to operate at vacuums up to 25 in., while the system at 2 Beekman Place was installed under the usual specification for vacuum systems, namely ability to circulate steam at 1 lb pressure with 7 in. vacuum on the returns. Such a specification does not necessarily insure a tight piping system, even though the terms are complied with under test.

It is a much simpler matter than is generally supposed to get a piping system, no matter how large, really tight and to keep it so. It would seem worth while to consider wording specifications in such a manner as will really insure tight systems, and the adoption of operating standards which will insure the maintenance of this tightness.

CHAIRMAN ROWLEY: If there are no other discussions on this paper, I will call on Dr. Hall to close the discussions.

DR. R. E. HALL: This report summarizes an investigation that was undertaken at the request of the New York Steam Corp. Our sole instructions were to examine conditions in steam and return lines and reach what conclusions we could. We were given complete leeway in the matter of experimental procedure. As you will note from the tables given in the paper, we were somewhat greedy for data, and caused much work getting samples and making analyses; but in the end we have been able to arrive at conclusions regarding requisite quality of steam for heating purposes, which, in view of the unavoidable conditions in the return system on the one hand do not penalize the steam producer by requiring unnecessary refinement in generating the steam, and on the other hand do not penalize the building owners because of fostering corrosion in his lines.

As Dr. Speller very definitely brought out, the corrosion in the return lines occurs at rather specific points and not generally through the lines where drainage is properly taken care of. I would like to emphasize this point, because it leads to the conclusion that it is the development of specific conditions at these particular points which is causative of corrosion. For instance, following a radiator corrosion product often collects in the traps or the nipples following them. This product is mainly iron oxide, and it collects whether the nipples are of iron or brass. It is necessary to conclude that the corrosion occurs in the radiator and the product of corrosion is then transported to the nipples. It is necessary so to conclude because in our examination of lines continuously under steam pressure we have never found corrosion. The path which the steam takes through the radiator, and whether or not it entraps the air present instead of forcing it out through trap or vent, may decide the occurrence or absence of corrosion. If in its passage through the radiator the steam drives out any air collected therein, corrosion does not occur; but if it entraps the air, corrosion is certain to occur. Thus, responsibility for the corrosion resides in the inherent characteristics of the radiator and not in the quality of steam provided by the generating station.

Following hot water heaters is another spot at which corrosion frequently occurs, and usually the channeling type of corrosion, or as Dr. Speller called it, the acid type of corrosion.

I must take issue with the characterization of the channeling type of corrosion as acid corrosion. An acid water may cause corrosion of the channeling type, but said type may be caused as well by flowing water in contact with oxygen and totally

devoid of carbon dioxide or other acids. Much confusion has been wrought by the common assumption that channeling corrosion in the return lines *ipso facto* must be due to the carbon dioxide present therein, regardless of its quantity. In the cases coming under our own observation we have in every instance found a free and easy path for the entrance of oxygen at the time when the thermostatic valve shuts off the steam pressure from the heater and permits a vacuum to develop. It is doubtless the contact of this high concentration of oxygen with the freshly formed and exceedingly pure condensate developed by the operation of the heater, that most frequently results in the channeling type of corrosion.

In a case under consideration at the present time, the return lines from the heaters are vented directly to the roof. Thus there is an open air connection through the return lines and the heaters to the steam valves. The correction of the condition in this instance is best effected by the use of a non-corroding metal for the relatively short return line connecting heater to the main return line.

A more careful study of the report before they wrote their discussion might perhaps have been beneficial to Messrs. Owens and Einert.

For instance, they state— "We gather that 20 ppm of carbon dioxide and 0.3 cc of oxygen per liter may be considered normal limits for feedwater and condensates." It was impossible for them to gather that idea from our report. What we said was, that *steam* containing 20 ppm of carbon dioxide and 0.3 ml per liter of oxygen was of satisfactory quality, and this is a totally different statement from that of the critics. Their statement would not have been in error if they had omitted "and condensates" from their criticism, but inclusion thereof leads them into criticism and discussion without bearing on the report we have presented. One of the most emphasized features of the report is that the concentration of extraneous components in the steam must not be confused with that in the condensate, for confusion of these two obscures any clear perception of the problems involved, and obviates any possibility of arriving at correct conclusions.

As a second example, they state that under the heading of Quantity Factor-Dissolved Oxygen, "we gather that the authors favor the ordinary type open heater." In this case again they gather what could not be gathered from our report, for we bluntly state in the report— "The preference of the authors is for (1) or (2) above, since either when operating properly acts as a governor providing a limiting concentration of oxygen in the system, so far as the generating system is its source." Our reference (1) was to the deaerating heater; (2) to the open direct contact heater with temperature maintained at 212 F and amply vented.

Unfortunately, throughout this discussion of Messrs. Owens and Einert there occur similar distortions by them of conditions and conclusions. We believe however that we have set forth the facts with sufficient clarity in the report to obviate necessity for pointing out further the specific misinterpretations.

We feel however that one further point is worthy of discussion.

Messrs. Owens and Einert state— "In the case of heating plants, inasmuch as corrosion in condensate lines is a major problem, we feel that the oxygen content should not be greater than that which can be removed with the best and latest type of deaerating heaters." On the other hand, the authors state— "Although the present-day technique of the well operated steam-generating station could readily and with certainty reduce the concentration of oxygen and carbon dioxide to a tithe of the value specified (0.3-0.5 ml/liter of oxygen and 15-20 ppm carbon dioxide), to do so is of no practical value so long as the steam passing from Division 2 (distributing system) of the system into Division 3 (utilization system) is degraded in quality by influx of oxygen from the inexhaustible reservoir of the atmosphere, and in amount dependent upon the physical and the operating characteristics of Division 3".

The position taken by Messrs. Owens and Einert follows naturally enough from

drawing conclusions while failing to recognize the difference in meaning of samples of steam taken by condensation in cooling coils, whereby all extraneous components are included in the sample, and the condensate formed in the system from and in contact with the steam, said condensate dissolving but minute fractions of the extraneous components. Conclusions based on such confusion are necessarily fallacious.

The authors appreciate the opportunity which this investigation has afforded to dissipate this confusion and to provide an economic basis for cooperation of steam producer and consumer without penalty to either.

No. 921

CHANGES IN IONIC CONTENT OF AIR IN OCCUPIED ROOMS VENTILATED BY NATURAL AND BY MECHANICAL METHODS

By C. P. Yaglou [1] (*MEMBER*), L. Claribel Benjamin [2] (*NON-MEMBER*), and Sarah P. Choate [3] (*NON-MEMBER*), Boston, Mass.

This paper is the result of research conducted at the Harvard School of Public Health in cooperation with A. S. H. V. E. Research Laboratory

INTENSIVE investigations have until now failed to discover the specific cause of *deadness,* or lack of a stimulating quality, in the air of occupied rooms, even when temperature and humidity are controlled, as contrasted with the air of the open country. Proponents of the *open air treatment* ascribe this quality of freshness to a vital principle which is lost when air is brought indoors, particularly when ventilation is effected by mechanical means.

In recent years, since the carbon dioxide, oxygen, and crowd poison theories have become obsolete, ionization has been suggested as the *air soluble vitamin,* but it has not yet been identified. The virtues of artifically ionized air have been extolled on purely theoretical grounds, with no scientific confirmation whatever.

The object of the present work is to study the problem of ionization in relation to ventilation and health. This paper deals largely with fundamental changes in the ionic condition of the air in occupied rooms ventilated by natural and by mechanical methods, and with the influence of various air conditioning processes upon the ionic content of air. Since the field is comparatively new, a brief discussion on basic principles of atmospheric ionization is also included.

Properties, Formation, and Destruction of Ions

According to Hess,[4] all gases, like electrolytes, contain positively and negatively charged carriers of electricity (*e.g.,* atoms, molecules, or molecular

[1] Assistant Professor of Industrial Hygiene, Harvard School of Public Health.
[2], [3] Technicians, Department of Industrial Hygiene, Harvard School of Public Health.
[4] See p. 2 of reference 1 at the end of this paper.
Presented at the 38th Annual Meeting of the American Society of Heating and Ventilating Engineers, Cleveland, Ohio, January, 1932, by C. P. Yaglou.

groups), which are called ions. The charge carried by an ion is the elementary charge; namely, 4.77×10^{-10} electrostatic units (E.S.U.). It is believed that this charge is the same in all gases and that it is equal to that carried by the hydrogen ion in the electrolysis of liquids. Because of this charge, ions move under the influence of an electric field and the direction which they take depends upon the sign of their charge.

In general, two classes of ions are recognized; the small or molecular size, and the large or Langevin ions. Owing to this difference in size, the speed which the two classes of ions attain in an electric field differs enormously.

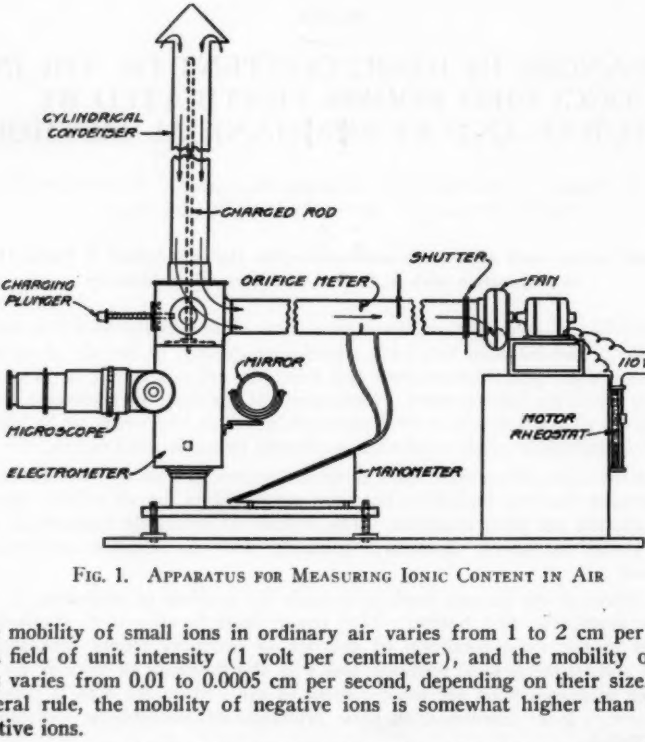


FIG. 1. APPARATUS FOR MEASURING IONIC CONTENT IN AIR

The mobility of small ions in ordinary air varies from 1 to 2 cm per second in a field of unit intensity (1 volt per centimeter), and the mobility of large ions varies from 0.01 to 0.0005 cm per second, depending on their size. As a general rule, the mobility of negative ions is somewhat higher than that of positive ions.

Large ions are formed by agglomeration of small positive or negative ions with condensation nuclei, such as dust, fumes, smokes, or drops of water. They are present in great numbers in city air which is polluted with products of combustion from chimneys and from automobile gases. Under such conditions the number of small ions is at a minimum, and it varies inversely with the number of large ions. Between the small and the large ions are the so-called *intermediate* ions, which are formed under certain conditions of humidity. Their mobility varies from about 0.1 to 0.01 cm per second in a unit field. Although

this classification is generally adhered to, experiments by the authors and by others [5] show that gaseous ions do not occur predominantly in any well-defined sizes, but that there is a continuous distribution of sizes from the very small to the very large. The frequency of distribution seems to depend largely on weather conditions and on the extent of atmospheric pollution.

In nature, ions are produced by solar radiation, by cosmic rays, and by radioactive changes in the soils of the earth.[6] Strongly ionized gases diffuse through the capillaries of the soil by the aspirating action of the wind and when the barometric pressure falls. It is believed that this soil respiration contributes about 60 per cent of the total ionic content of the air near the surface of the earth.

The state of atmospheric ionization is maintained, more or less, by the simultaneous actions of other natural processes which tend to destroy or to neutralize the ions. The most important of these are (1) recombination of ions of opposite charge to form neutral ions, (2) agglomeration with condensation nuclei to form large ions, and (3) diffusion and adsorption by solid or liquid conductors.

For a more thorough discussion on physical properties and theories of atmospheric ionization, the reader is referred to an excellent treatise by Hess.[6]

MEASUREMENT OF IONIC CONTENT OF AIR

The number of positive and negative ions in a unit volume of air is usually counted by means of an apparatus devised by Ebert. Fig. 1 shows the modified form of Ebert's apparatus which was used in the studies described in this paper. A stream of air is drawn by fan suction through a cylindrical condenser, the central rod of which is charged to a known potential with a polarity opposite to that of the ions to be counted. The charged rod is well insulated and it is connected to the quartz fibers of an electrometer. All other parts of the apparatus are grounded.

As the air passes down the condenser tube, ions of opposite sign are attracted to the rod, and upon striking it extract a quantity of charge equal to their own. Only those ions will reach the charged rod which have sufficient velocity (mobility) to carry them across the intervening space before they are carried away by the air stream. From the rate of discharge of the electrometer and the rate of air flow—the latter measured by an orifice meter—the number of ions per unit volume of air can be computed.

If $n\pm$ is the number of positive or negative ions in a cubic centimeter of air, w the volume of air passed through the condenser, v_0 the initial charge in volts, v_1 the final charge, corrected for natural leakage if any, e the charge carried by an ion (4.77×10^{-10} E. S. U.), and c the combined electrical capacity of the condenser and the electrometer in E. S. U., the following equation will hold:

$$(n\pm)\,ew = \frac{c\,(v_0 - v_1)}{300} \text{ and } n\pm = \frac{c\,(v_0 - v_1)}{300ew} \qquad (1)$$

[5] See references 2 and 3.
[6] See reference 1.

The division by 300 gives the loss of potential in E. S. U. The value of $n+$ is obtained by charging the system negatively, and that of $n-$ by a positive charge. Usually it takes five minutes to make an observation. Less time is required when the ionic content is unusually high.

The number and the size of ions caught depend largely upon the design of the instrument, the voltage charge, and the velocity of the air through the condenser. The instrument used by the authors was designed to catch all small ions having mobilities from the maximum possible down to about 0.2 cm. per second per volt per centimeter. The authors adopted this particular group of ions because it was found to be sensitive to weather changes and to changes in the general indoor atmosphere produced by respiratory and metabolic proc-

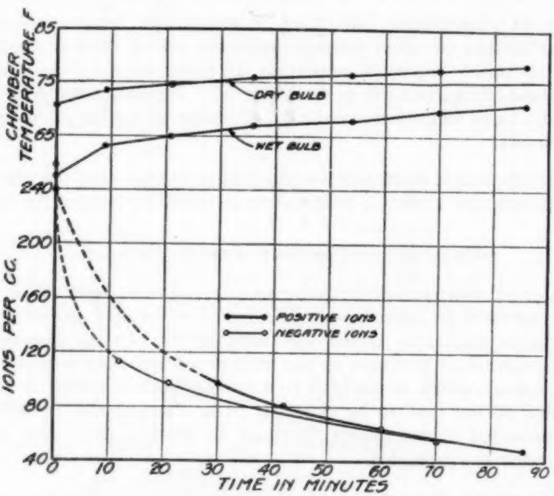


FIG. 2. INFLUENCE OF RESPIRATION AND TRANSPIRATION ON IONIC CONTENT *

* Seven Persons in a Closed Chamber. Cubical Contents, 550 cu ft. No Ventilation.

esses, and by indoor activities. Unless specifically stated otherwise, the terms *ions* or *ionic content*, as used hereafter in this paper, refer to this particular group.

By changing the condenser, increasing the voltage charge and reducing the air flow, the instrument used by the authors can be adapted to catch, in addition to the small ions, a large portion of the intermediate and a portion of the heavy ions, as is the case with the usual Ebert's apparatus. From the standpoint of ventilation, however, counts made under these conditions are uncertain, owing to the unknown action of the heavy ions which happen to be near the charged rod of the instrument. When the number of large ions is great, as on dusty and foggy days, those near the charged rod are likely to be caught accidentally, thus vitiating the results. In the apparatus used, the conditions necessary for capture of the small ions are not appreciably exceeded, and

therefore the number of large ions caught accidentally is usually very small. When thick dust clouds are raised, however, either by unusually strong winds or by vigorous sweeping and dusting indoors, an appreciable number of large ions are caught accidentally. The only alternative at present is to discard such readings. This is an unfortunate instrumental limitation which the authors hope to correct. In addition to the sources of uncertainty just mentioned, the whole technic of securing accurate results is intricate and laborious, even with the improved apparatus.

Daily measurements (since May, 1930, Sundays excepted) of ionic content of indoor and outdoor air disclose very definite seasonal trends which will be discussed more thoroughly in a subsequent paper. For the purpose of this paper it is sufficient to state that atmospheric ionization undergoes great diurnal and seasonal variations, depending upon local and general meteorologic conditions. It is much higher in summer than in winter, much higher on clear days than on rainy, foggy, or grey days, and, as a general rule, higher in the day time than at night.

The maximum variation in the particular group of small positive or negative ions which were studied (down to a mobility of 0.2 cm. per second) is from about 50 on grey winter days to about 700 on clear days in summer. In contrast with general assumptions, the ionic condition in unoccupied or very lightly occupied rooms does not differ greatly from that out of doors and sometimes it may even be higher, as will be shown later. However, a few people in a room may alter the situation, even though the amount of air space per occupant is much greater than that which is considered adequate in modern ventilation practice.

INFLUENCE OF RESPIRATION AND TRANSPIRATION[7] ON IONIC CONTENT IN OCCUPIED ROOMS

In order to study the influence of room occupancy on ionic content, three different series of experiments were conducted. In the first series, seven persons were seated comfortably in an air tight steel chamber (experimental compression chamber, cubical contents 550 cu ft); readings of ionic content were taken both before they entered and continuously while they were in the tank. Smoking was not allowed inside the chamber.

It can be seen from Fig. 2 that both positive and negative ions decreased from an initial concentration of about 250 ions per cubic centimeter to one of about 50, after the people had been in the chamber for 85 minutes. The decrease was very rapid during the first 20 minutes; after that it was gradual. A duplicate experiment, with no one in the chamber and with the ion counter outside, showed no apparent change in the number of ions with time, so that the entire effect must have been due to the occupants and, to some extent, to the de-ionizing action of the ion counter itself. At the beginning, the rate of fall in the number of negative ions was greater than the rate for positive ions; this was probably due to the higher mobility, and hence higher diffusion, of the negative ions to conducting surfaces.

In another series of experiments, ion counts were taken in ordinary classrooms and in reading rooms, under conditions which were fairly representative

[7] Exchange of gases through skin and clothing.

of those in lecture halls ventilated by the window-gravity methods. Most of these observations were made in a classroom, 30 ft by 33 ft by 10 ft 6 in., having four windows on a northeast wall and three windows on a southeast wall, the window area being about 20 per cent of the floor area. A double door from the room opens directly into the main corridor.

The data shown in Fig. 3 were obtained in this classroom during a luncheon talk in which 34 persons ate luncheon and smoked, while listening to a lecture on current research. No attempt was made to control the general room

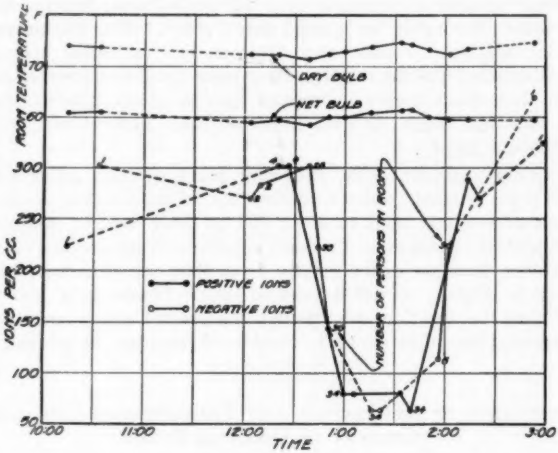


FIG. 3. INFLUENCE OF ROOM OCCUPANCY ON IONIC CONTENT [a]

[a] Room Dimensions 30 ft x 33 ft x 10 ft 6 in. Number of Occupants, Thirty-four. Good Window-Gravity Ventilation.

conditions; the occupants were at liberty to open or close windows and doors, and to come and go as they pleased.

In spite of what might be considered good ventilation (see small temperature rise in Fig. 3), both positive and negative ions fell from about 300 to 65 ions per cubic centimeter in approximately 20 minutes after the people entered, and they remained at this low level until the occupants left the room. It took nearly an hour for the ionic content to resume its initial value after the people departed, all other conditions remaining unaltered. The minimum ionic content in Fig. 3 probably represents the irreducible minimum, due to the influence of radio-active substances in the plaster of the walls and to cosmic rays, which are capable of penetrating thick metal walls and ionizing the air of enclosed spaces.[8]

These and many other similar data show that there is a most striking change in the air of occupied rooms as the result of a reduction in ionic content. It makes little difference whether a room is ventilated by natural or by mechanical

[8] See reference 1, p. 117.

methods. The problem now is to determine whether such alterations in the electrical quality of air have any significant bearing on health.

The loss of ions in these experiments cannot be accounted for by respiratory processes alone, even if it is assumed that expired air is completely devoid of ions. This is because the volume of air breathed by the occupants (0.28 to 0.42 cfm per person) is very small in proportion to the cubical content of the room (300 cu ft of air space per occupant in the classroom). The de-ionizing action of the ion counter itself is also small (air flow 5 cfm). Transpiratory processes through skin and clothing may account for a considerable portion of the loss, but the problem seems to be complicated by the effects of tobacco smoke, food, and possibly other factors. More data are necessary in order to derive an ionic balance.

OUTDOOR AIR SUPPLY IN RELATION TO IONIC CONTENT

The third series of experiments on the effect of room occupancy on ionic content was carried out in an air-conditioned room, where it was possible to measure the quantity of outdoor air supplied per person per minute in order to maintain normal ionic content. This was done by placing twenty-four persons (men and women seated in armchairs) in the conditioned chamber, and measuring the ionic content at various rates of air change, as shown in Table 1.

Since it was a warm day, the air was cooled to a fairly comfortable temperature by means of pipe coils through which cold brine was circulated. For reasons to be discussed later, the dehumidifier was not used in cooling. When the subjects entered the room, the fans were stopped, and the ionic content was allowed to fall to an irreducible minimum level. The fans were then started again, and were kept running at constant speed until a new equilibrium was established. The procedure was repeated with gradually increased air supply up to the maximum capacity of the fans.

Table 1 gives the experimental conditions and the results secured. A better idea of the relationship between outdoor air supply per occupant and ionic content in the room may be obtained from Fig. 4, which indicates that a ventilation rate as high as 160 cfm per person is barely sufficient to maintain normal ionic content. The ionic content does not appear to be significantly higher with the usual air supply of 30 cfm than with no ventilation. This is consistent with experience in classrooms which were fairly well ventilated by natural methods (see Fig. 3). The minimum ionic content in Fig. 4 appears to be higher than that in Fig. 3, perhaps because no smoking was allowed in the former case when the fans were shut down. When the fans were started again, smoking was quite general among the subjects.

ARTIFICIAL IONIZATION

From the results of the foregoing experiments, it is quite evident that if the ionic content of air has any beneficial effects on health, some artificial method for ionizing the ventilating current must be employed.

The last two lines in Table 1 give data secured under conditions of artificial ionization. An ionizer was adjusted to produce about 5,000 small positive ions per cubic centimeter of room air (calibration with 2,050 cfm and one person in room) and the least possible number of negative ions, 635 in this case. When there were 24 persons in the room and there was an air supply of 85 cfm

TABLE 1. OUTDOOR AIR SUPPLY IN RELATION TO IONIC CONTENT

Number of Persons in Room	Ventilation Rate (cfm)		Small Ions per cc			Condition of Air				Remarks
						Supply Registers		Exhaust Registers		
	Total	Per Occupant	Positive ($n+$)	Negative ($n-$)	$\frac{n+}{n-}$	Dry Bulb (Deg. Fahr.)	Wet Bulb (Deg. Fahr.)	Dry Bulb (Deg. Fahr.)	Wet Bulb (Deg. Fahr.)	
2	2050	1025	343	340	1.0	76.0	58.0	76.8	58.4	Initial readings before occupants assembled. All air from out of doors. Cooling by means of central fan system.
24	0	0	95	96	1.0	...	...	83.0	71.2	Fans off. Natural air leakage through dampers and crevices. Strong body odors. Minimum ionic content reached in 36 minutes after occupants assembled.
24	2050	85	184	129	1.4	76.0	57.8	80.0	60.5	All air from out of doors. Cooling system in operation. Body odors just perceptible to sense of smell upon entering room.
24	2550	105	207	160	1.3	75.3	56.9	79.2	58.8	All air from out of doors. Cooling system in operation. Body odors just perceptible to sense of smell upon entering room.
24	3450	145	345	303	1.1	77.8	58.0	79.5	59.2	All air from out of doors. Cooling system in operation. Body odors not perceptible to sense of smell on entering room.
24	2050	85	2790	80	35.0	75.0	56.5	78.8	58.5	Artificial ionization. All air from out of doors. Cooling system in operation. Body odors not perceptible to sense of smell on entering room.
1	2050	2050	4810	635	7.6	...	...	...	...	Artificial ionization. All air from out of doors. Cooling system in operation. After occupants departed.

per person, the number of small positive ions was increased from 184 to 2,790 per cubic centimeter of air, and the negative ions decreased from 129 to 80. Probably, the difference between the output of the ionizer and the actual ionic content of the air in the room represents the loss by respiration and transpiration.

This and other experiments on artificial ionization indicate that it is fairly practicable to control the ionic content in occupied rooms at any desirable level, up to a maximum of 1,000,000 ions per cubic centimeter, without producing a perceptible quantity of ozone. According to the physiologic experiments of the authors, it is doubtful whether a concentration higher than 2,000 ions would be needed in ventilation work, and experiments are now in progress to determine the threshold value. The maximum ionic content that the authors have recorded in Boston, since May, 1930, has been about 700 ions per cubic centimeter, in clear summer weather. Subjective sensations in artificially ionized air, together with effects on health and physiologic reactions, will be discussed in another paper.

INFLUENCE OF AIR-CONDITIONING METHODS ON IONIC CONTENT

There seems to be a growing belief that modern systems of mechanical ventilation deprive the air of its ionic content, by diffusion and adsorption to grounded metal surfaces, and by overheating. Experiments in our psychrometric room failed to confirm this assumption, except when the air was brought in contact with a fine water spray by passing it through the usual type of dehumidifier.

The influence of various air conditioning processes was studied, both separately and in combination, by taking ion counts in the psychrometric room before and after making similar observations out of doors at the air intake. In this way, diurnal changes in ionic content were accounted for, to some extent; but the compensation was never entirely satisfactory, owing to rapid diurnal changes. Therefore, for the time being, discussion shall be confined to general tendencies only, until a second ion counter is secured for making simultaneous measurements indoors and out of doors.

In the air-conditioning installation used in these experiments, the air supply is taken from an open yard, at a height of about 7 ft above the ground. The air travels a distance of about 100 ft, through a 17 by 22-in, galvanized-iron duct, before it reaches the inlet registers in the psychrometric room. On its way to the room, it passes through seven 90 deg elbows, a dehumidifier (3 ft $\times$ 3 ft $\times$ 9 ft), six stacks of heaters, or coolers, in series, and two louvre dampers. In the particular experiments which were performed for the purpose of determining the effect of air conditioning processes, there was one person in the room, and the air circulation through the system was 2,050 cfm. This is the minimum air supply with dampers wide open.

With simple ventilation (no air conditioning), the overall loss in ionic content, from the outdoor air intake to breathing zone level in the psychrometric room, varied from 0 per cent to 10 per cent for the positive ions, and from 0 per cent to 30 per cent for the negative ions.

In contrast with the prevailing belief, heating the air by means of heating units used with central fan systems increased both positive and negative ions

in all experiments. As a general rule, the increase was sufficient to counteract the loss by diffusion and adsorption to metal surfaces, and in a few instances, the ionic content of the room air was appreciably higher than that out of doors. Cooling, on the other hand, decreased appreciably both positive and negative ions.

These effects of heat and cold are in accord with a well-known connection between temperature and atmospheric ionization.[9] The fact that in cold weather the ionic content in unoccupied heated rooms is often higher than that out of doors, may be explained in part or in whole by this temperature relationship. A good circulation of raw outdoor air through open windows is instrumental in quickly reducing the ionic content in the room to the outdoor level.

The most striking effects of air conditioning were produced when the spray

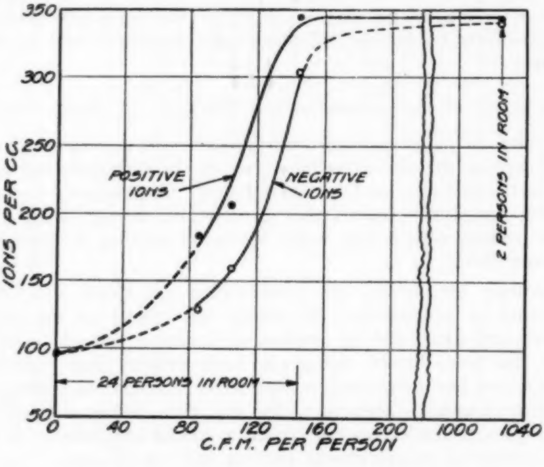


FIG. 4. OUTDOOR AIR SUPPLY IN RELATION TO IONIC CONTENT

system of the dehumidifier was in operation. Whether the air was washed. humidified, or dehumidified, it was deprived of all small ions having mobilities upward of 0.9 cm per second. Moreover, the spray produced a great number of large negative ions, or condensation nuclei, similar to those produced by hard rain (*Lenard effect*[10]), smoke, or fumes. The higher the water pressure at the sprays, and the dirtier the spray water, the greater was the number of large negative ions. Large positive ions were also present but in comparatively small numbers. Since the instrument was designed to count small ions, it was impossible to catch more than 20,000 intermediate and large ions per cubic centimeter. With all instrumental modifications possible, the computed minimum mobility of the ions caught was 0.03 cm per second, as compared with the minimum known value of 0.0005 cm per second for the largest ions.[11]

[9] See reference 1, p. 46.
[10] See reference 1, p. 63.
[11] See reference 1, p. 8.

Simple recirculation (no air conditioning) gradually reduced both positive and negative ions, and when there were more than three persons in the room the irreducible minimum was substantially the same as that in Fig. 2. In all cases the negative ions were affected more quickly than the positive ions, as is to be expected from their higher mobility.

SUMMARY

A series of experiments was carried out in rooms, both occupied and unoccupied (1) with no ventilation, (2) with window-gravity ventilation, and (3) with mechanical ventilation, in order to determine the extent to which the number of small ions is affected by respiration and transpiration, and by modern air-conditioning methods.

In contrast with the prevailing belief, the ionic content in unoccupied heated rooms did not differ much from that out of doors, and in cold weather it was often higher, owing probably to a temperature effect.

In occupied rooms there was a marked decrease in both positive and negative ions. Immediately after the occupants assembled, the ionic content of the air fell abruptly to a very low value, which was maintained until the occupants left the room. Both positive and negative ions began to rise again as soon as the people departed.

The minimum supply of outdoor air required to maintain normal ionic content in a crowded room was found to be prohibitively high (160 cfm per person). With the usual air supply of 30 cfm per person, the ionic content did not seem to differ greatly from that with no ventilation at all. On the other hand, it was possible by means of artificial ionization to control both the quantity and the quality of ions at any desired concentration up to 10,000 ions per cubic centimeter, with or without ventilation.

Mechanical ventilation reduced the ionic content from 0 per cent to 30 per cent by diffusion and adsorption to metal conductors. Heating the air by means of a central fan system increased the ionic content, and cooling by similar methods decreased it. The usual methods of washing, humidifying, or dehumidifying by means of water sprays, deprived the air of all small ions, and produced a great number of large negative ions, or condensation nuclei, by the well-known *Lenard effect*. Recirculation reduced both positive and negative ions by diffusion and adsorption to metal conductors.

REFERENCES

1. Hess, Victor F.: *The Electrical Conductivity of the Atmosphere and its Causes.* New York: D. Van Nostrand and Company, 1928.
2. Busse, W.: *The Size Distribution of Ions in Gases.* Ann. Physik, 1926, *81*, 262.
3. Laporte, Marcel: *A Critical Study of Methods for the Determination of the Mobilities of Gaseous Ions.* Ann. Physik, 1927, *8*, 710.

DISCUSSION

ELLSWORTH HUNTINGTON[12] (WRITTEN): We have long known that there was some condition of the atmosphere aside from temperature, humidity and movement

[12] Prof. of Geological Science, Yale University.

which seems to have an influence upon human comfort and health. We have suspected that this condition might be the electrical condition of the air. I have tried a number of times to find some way of investigating the matter but have never had the proper facilities. Mr. Yaglou's work does not prove that the ionic content of the air is the factor for which we have been searching. It does, however, indicate that at last we are on the way toward a definite solution of this particular problem. I hope that he will be able to make a long series of experiments in which he changes the ionic content of the air and observes the effect upon people's feelings. I also hope that he is going to measure the ionic content of the outer air, day after day, and compare this with fluctuations in health. His work is so promising that it deserves full support.

W. J. McCONNELL,[13] M.D. (WRITTEN): The supposition that some "vital principle" of the air is lost when air is brought indoors as is evidenced by stuffiness or deadness referred to by the authors of this paper, even though the physical qualities of the air are controlled, has advanced beyond academic interest and threatens to modify our present conception and practice of air conditioning.

Whether this uncomfortable and stuffy feeling of which most people are sensible, can be attributed to a change in the ionic content of air or is produced by the longer infra red rays acting on the skin, and reflexly through the nerves on the mucous membrane of the nose, as suggested by Sir Leonard Hill in a recent lecture delivered at the Royal Institute of Public Health in London (and published in its journal for December), or to some other as yet unidentified quality of the air, remains to be determined.

There appears to be very little evidence so far adduced to support the practice of artificially ionizing air, and it is gratifying to learn of this well planned and comprehensive investigation to determine the real value of ionization in relation to ventilation and health. The authors have related in this paper the results of well designed experiments to demonstrate the changes in the ionic content occurring in the air of confined spaces under varying conditions. They have removed the problem of ionization from the realm of mysticism to one of reality.

An evaluation of artificially ionized air following the series of experiments which the authors state are now in progress with the intention of determining the threshold value of ionization and its relevant bearing on health is earnestly anticipated.

W. RAY MONTGOMERY (WRITTEN): As our knowledge has grown about the ions in the air, the relation between atmospheric ion content and air conditioning has become of increasing importance. It is common knowledge that normal air contains an abundance of both positive and negative ions. It is also known that these ions vary in number according to position on the earth's surface and to the time of day and season.

What effect these atmospheric carriers of electricity have upon human life through our breathing and being in continual contact with them is the problem ventilating engineers now face.

As early as 1923 our firm publicly introduced the thought "Has air a vital property?" It was then suggested that in ionization would possibly be found the air soluble vitamin. In 1929, in a paper delivered by Frank E. Hartman, it was pointed out that ionization is in no sense to be considered a panacea in ventilation. It is a factor and a very essential factor that sooner or later must demand recognition.

During the past two years, or since 1929, I have studied the operation of over a hundred mechanical ionizing machines which are used in conjunction with air conditioning installations throughout the United States.

---

[13] Assistant Medical Director Metropolitan Life Insurance Co.

Reports from these jobs based upon the opinions of the occupants of these buildings prove conclusively that ionization in its relation to ventilation is definitely established. It is quite true that these opinions are based entirely upon sense impressions but is it not also true that the final test of the air condition in any system resides in the sense impression, since it is entirely the sense impression that air conditioning proposes to satisfy?

Professor Yaglou's work shows convincingly that the presence of human beings tends to rob the atmosphere of its ions. Just how the human body consumes the ions it removes from the air is still a matter for further research. Undoubtedly a good many of them are taken up by the lungs during respiration and it would seem that the skin is responsible for the removal of others, and while Professor Yaglou's results do not give specific data on the effect of ions upon health, still it is significant that human beings are responsible for the removal of ions from the air in ventilated spaces.

Recently, Dr. Frederick Dessauer, Director of the Institute for the Physical Elements of Medicine, University of Frankfurt-am-Main, Leipzig, published in book form a series of fifteen original researches in commemoration of the 10th anniversary of the foundation of the Institute. In this work which we understand covers the 10-year period and which is devoted almost entirely to the therapeutic values of ionization, it is pointed out conclusively that the relation in numbers between the positive and negative ions has a direct bearing upon subjects which are suffering from certain types of disease.

Referring again to Mr. Hartman's article of 1929, evidence was presented therein which unquestionably points to ionization as essential to air conditioning in regard to both number and polarity of the ions.

In view of Professor Yaglou's most interesting results, the unquestionable and reliable work of Dessauer and the favorable reports received on the installations throughout the country where mechanical ionization has been used, for a period of two years, we are of the firm conviction that ionization in relation to ventilation is just entering upon a broad period of recognition and commercial development. The field is by no means thoroughly explored and the problems should be intensely investigated by the Research Department of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS to establish it upon a firm foundation.

Certainly, we feel that in the relation and number and size of positive to negative ions in air conditioning, will be found the answer to both health and comfort.

FRANK E. HARTMAN (WRITTEN): This paper, of Mr. Yaglou, and his associates, is indeed interesting—not only for the material which it contains, but also in its timeliness, for the subject of this work is very much at issue, at this time, in Europe, and promises to become the centre of considerable discussion, and, doubtless, the stimulus to much research work here in America.

As instances of this, several months ago the *Journal of Diseases of Children* devoted the entire of its section II to contributions concerning the effects of climate. Only last month the subject of "stuffiness" in heated and ventilated buildings formed the basis of a symposium at the Royal Institute of Public Health, at London, whilst the December 4 issue of the *Deutsche Medizinische Wochenschrift* is devoted, in the main, to what Dr. H. Linder, Chief of the Sanitarium Bella Lui, at Montana, Canton Wallis, has aptly termed "bio-climatic affairs."

My own interest in the subject dates back some 10 years, as the members of our Society will doubtless recall. During that time I have devoted considerable time to its study, hence this paper has more than a passing interest for me, and I want to add that I can well appreciate the patience and painstaking efforts required to present the data which Mr. Yaglou and his associates have given us.

I can appreciate the difficulty of selecting a single piece of apparatus with which to explore the field of ionization, since to encompass the entire range of ionic mobilities, with a view of determining their separate number, quite an array of apparatus appears to be, and is, in fact, essential.

However, Mr. Yaglou states that their concern was, in the main, with small gas ions. In such event it would seem that the Ebert apparatus, as modified by Swann and later developed by the Carnegie Institution of Washington, would more conveniently answer the purpose. This apparatus has been highly refined by the Carnegie Institution, however, when we found it necessary to purchase this instrument we were forced to send to Germany for its manufacture, although the drawings and specifications were furnished us from Washington, D. C.

Because of the increasing interest in atmospheric electricity, as related to ventilation, that will follow naturally from Mr. Yaglou's experiments, I believe that a few details concerning apparatus will not be out of place here. Points of difference between the Swann modification and the conventional Ebert apparatus, as illustrated in Mr. Yaglou's paper, lie both in the measuring condenser and electrometer. The conventional apparatus employs a bifilar Wulf electrometer, which requires potentials of the order of 100 volts on the collecting system. This increases the insulating problem, which is sufficiently difficult even with much lower potentials. In the Swann modification a single fiber electrometer, which is many times more sensitive, is employed. This permits of potentials on the collecting system of the order of hundredths of a volt. With this electrometer leak tests are frequently of nil value.

The method of capturing the ions, and in consequence the construction of the measuring condenser, differ appreciably between the modified and conventional forms. In the modified apparatus two concentric cylinders are employed, and insulated from each other. The outer cylinder is earthed and serves to protect the condenser from the action of stray fields. The inner cylinder is charged up to a potential of from 90 to 100 volts, of the same sign as the ions to be captured.

A rod is located at the centre of the inner cylinder and connected with the fiber of the electrometer. The approach of the rod (*i.e.* the end facing the incoming air) is protected by a shielding tube, which serves to prevent incoming ions from being repelled from the condenser through the action of the field on the inner cylinder.

The ions pass between the inner cylinder and the rod, precisely as in the conventional apparatus, however, in the case of the modified form the field on the inner cylinder drives ions of like sign over to the rod, where they are deposited thus increasing the tension of the rod, and this increase is indicated by the electrometer.

It is immediately seen that the problem of capturing large ions, when present in great numbers, is nothing like so great, if it exists at all, with the modified apparatus as with the conventional form. In the modified apparatus the potential on the collecting system is always low (of the order of several hundredths of a volt), thus its attractive influence on large ions is negligible. The high potential, serving as a driving force for the small ions, is confined to the inner cylinder, too remote from the collecting rod to permit the influence of the driving field to deposit large ions in the period of time that unit volume of air remains in the condenser. Of course some large ions of opposite sign are collected on the inner cylinder, but since the potential of the cylinder is maintained by a battery, directly connected on, this causes no change in driving potential during a count.

Mr. Yaglou and his associates remark: "The problem now is to determine whether such alterations in electrical quality have any significant bearing on health." This is indeed a problem, the solution of which will require much painstaking work, and of a far-reaching and complicated nature.

One phase of this problem, that has troubled me for some time, is just where, and how, to distinguish between the effect of change in electrical quality and elec-

trical quantity, or better put, effective space-charge-density, resulting from the sum of all ions present.

It is well to bear in mind that the curves presented in this paper show changes in one, and numerically speaking, frequently small group of ions, and do not at all tell us what happened to these highly mobile ions. Perhaps as ions they were not all lost, but rather became captured by condensation nuclei or dust particles, thus merely passing from the highly mobile class to classes of lesser mobility. In this event the total charge of the air would not have been appreciably altered, and this brings us face to face with the necessity of distinguishing between the effect of total charge-magnitude and group charge-magnitude, if I may coin the phrases, of the many classes of ions.

Three years ago, in cooperation with Dr. G. R. Wait, of the Carnegie Institution of Washington, we made ion counts in the class rooms of several schools. At that time we determined the number of small ions, together with the total conductivity, as determined by the Institution's modification of the Gerdien's apparatus, and also the condensation nuclei, as determined by the Atkin counter. One thing that we found very definitely at that time, and which I have recently confirmed, is that upon occupancy of a room by people, the number of condensation nuclei increases tremendously, so greatly, in fact, as to become uncountable with apparatus available, although ordinarily adequate for meteorological observations. Now these nuclei will capture small ions. Clothes carry and distribute dust in the air, smoking produces myriads of particles, these too capture ions. In fact small ions must disappear quickly from occupied rooms.

Due to high mobility of small ions they produce a very definite influence on conductivity, in fact, under most conditions, conductivity is the result, largely, of the presence of small ions. In cases where we measured total conductivity simultaneously with counting small ions, the loss of small ions showed an influence on conductivity, but since the large ions were not separately determined it is difficult, if not impossible, to arrive by calculation at any conclusion as to whether the small ions were entirely lost or simply converted into large ions. We have been contemplating a set-up wherein the air will pass in series through both small ion and large ion counters, with the view of throwing some light on this point. The counting of large ions is very tedious. The readings are long. Insulation leak is very annoying and frequently difficult of location and mending. Measurements of total conductivity are much simpler, but leave one with data that are difficult of utilization.

Early in 1931 the results of 10 years' work on the physiological influence of air ions was published in Germany, and these results would convince one of an important significance of atmospheric electricity, especially in ventilation. However, only last month Sir Leonard Hill, in his address to the Royal Institute of Public Health, attributes stuffiness "not to a mystic quality of ionization," to quote the *Lancet*, "which is sometimes invoked, but to the absorption by the horny epidermis of the long infra-red rays emitted in relatively large proportions by radiants, such as dull red electric fires. . . ."

The German work to which I have just referred is that of Prof. F. Dessauer and his associates, at the Institute for Physical Foundations in Medicine, of the University of Frankfurt-am-Main. According to Dessauer small ions have little if any physiological effect. The reasons given for such conclusion, however, are not exactly patent. In his experiments subjects were supplied with air containing excessively large numbers of small ions, of practically one sign (in some cases positive and in others negative), however, in the exhaled air no small ions were found. From this it was concluded that small ions, being highly diffusable, became adsorbed on the buccal membranes, the trachea and, perhaps, the bronchii, but did not penetrate deeply into the lungs proper. In cases of subjects being supplied with like quantities of

large ions, some proportion of these large ions was always found in the expired air. The dismissal of this question on the basis of the foregoing seems frought with possible error, since nothing is said of the possibility of the small ions being converted into large ions by the nuclei present in exhaled air. Rather the conclusion should be based on physiological findings, and certainly Prof. Dessauer and his associates report ample physiological evidence in support of their favorite "dust-ions."

In this work, reported on by Dessauer, et al., considerable attention is given to charged particles of MgO, or, as they call them, "dust-ions." The mobility of these ions lies between 0.007 and 0.0018 cm, corresponding to radii of from 100 to 190 x 10^{-8} cm. Treating subjects, and particularly "pathogenic cases," with magnesium oxide ions of negative polarity, some very definite results are reported. A sense of freshness in the air is ascribed to them, also a stimulating effect, as well as a definite lowering of blood pressure in cases of hypertension. On the other hand similar treatment with ions of positive sign caused increase in blood pressure, and in some cases a feeling of fatigue was reported. This is an interesting observation, particularly in view of Mr. Yaglou's findings regarding the effect of washing air with water sprays, producing a great number of large negative ions. This, from Dessauer's work, would seem desirable, and may account, in part, for the *"freshness"* frequently ascribed to washed air.

In working with artificial ionizers, we believe, that it is highly important that the ionizer have no effect on the air other than one of ion production. I am referring now to research work, directed towards a study of the effect of ions per se. If the ionizer has some chemical effect on the air it is going to be very difficult to isolate it and draw the line between the two forces. The writer, working in cooperation with Mr. Sam Bloom and Dr. Mark Jampolis of Chicago, several years ago, obtained some very definite effects on laboratory animals, using a source of artificial ionization. We attempted the duplication of this work employing minute traces of ozone, below the threshold of olfaction. The results were very conflicting. We did get catarrhal symptoms and pneumonias in some cases with both the ionizing apparatus and the ozone apparatus. Of course small animals are much more susceptible than humans, especially to ozone. We do know that the degree of ionization was small with the ozone apparatus, since only an exceedingly small quantity of air was passed through the ozonizer, and admitting it to be highly ionized, upon dilution with the air supplied the cages, the ion number would be small. That our results were complicated by some chemical effect on the air the writer feels confident, and since it is easy to produce ions under conditions that at least tend towards an absolute minimum of accompanying chemical effect, the writer feels that this distinction should be made in more purely research work.

In closing I want to express my appreciation of the work reported by Mr. Yaglou and his associates. And I want to congratulate the Society on their far-sightedness in supporting this work. When one considers that the lungs present about ninety times the surface presented by the epidermis, the importance of our air environment cannot be over-estimated. This, together with the present mechanistic trend of science, and the invasion of the physiological and biological sciences into the fields of physics, promises to make atmospheric electricity an important study of the immediate future.

J. I. LYLE (WRITTEN): It is interesting to note that ionization is as much of a mystery to a broad cross-section of the people of this country as humidity was 15 years ago. A contributing reason for this is, undoubtedly, that ionization is more difficult to observe than humidity, and that its effect, if any, upon human beings is more subtle.

Considerable progress has been made in the development of equipment for causing ionization, but it is still a difficult and tedious matter to observe the extent of ioniza-

tion. If we are to apply it successfully we must develop a simple method of determining the ionic content of air and, eventually, of automatically regulating that content within reasonable limits.

It is interesting to note from Figs. 2 and 3, another instance in which the massing of people creates its own climate, a fact which has been long recognized by those working with air conditioning equipment. The sudden change brought about by the massing of people shows that in our present stage of development, ionizing equipment should be applied individually to rooms, rather than used as a central station equipment. Furthermore, unless excessive air quantities are used, crowded spaces can not be brought up to a normal ionic content by central station equipment.

We know that it is essential to counteract the effect of masses of human beings upon the closed spaces which they occupy by means of air conditioning, and we know the results we are able to obtain are relatively good, but we are still guessing at the economic and health value of air conditioning and ventilating. A vast amount of work will be required for this, and the cost of it will be too great to be borne by any single commercial enterprise. The practical way seems that of enlisting aid by obtaining funds from some wealthy source and of carrying on the work by combining the interests of the air conditioning industry with that of public health.

M. J. Rosenau [14] (Written): The study of the air in its relation to health is an intriguing subject. Many theories have been advanced to account for the ill effects of "bad" air in a crowded, ill-ventilated room.

The first scientific study of this problem was by chemists who failed to disclose any evident relationship between the chemical changes in the ordinary components of the air and "crowd poisoning." The school of Flügge revealed that the problem is physical rather than chemical, and closely bound up with the physiology of heat loss from the body. In accordance with this theory, the fact was confirmed in many laboratories everywhere that the major problem in ventilation is air conditioning with special reference to temperature, humidity and moisture. This entirely revolutionized our notions of ventilation and the effect of air upon our well-being. Many students of the subject jumped to the conclusion that it solved the entire problem. There remains something, however, that is still undetermined, somewhat mysterious, concerning the air that science has not reached. Therefore, the studies of the ionic content of the air by Yaglou, Benjamin and Choate have special significance, and I hope very much that this work will be driven further now that the foundation has been laid to determine what effect if any the varying ionic content of the air we breathe has upon our health.

F. C. Houghten: Anyone who has studied the relation of atmospheric conditions to the health and comfort of man during the past 10 years has looked for the discovery of some factor which would have an important effect in ventilation. Some have called this unknown factor the vital characteristic of the air, or some magic quality which would change in respect to its health and comfort-giving qualities.

The search for such a quality has paralleled the finding of a similar quality in food, known as vitamin. We used to think carbohydrates, starches, fats and proteins were all the important qualities to be found in food. Then vitamins were discovered to be a necessary quality. It has since been shown that vitamin in food is at least somewhat related to ionization; possibly, it is ionization.

In searching for a vital characteristic in the atmosphere, a great deal of attention was at first paid to ozone. Many physiologists thought of ozone as being the vital characteristic. Later, it was thought that ultra-violet radiation was the sought-for quality, but neither ozone nor ultra-violet radiation has satisfied the hopes that were aroused.

---

[14] Harvard University Medical School.

Now Professor Yaglou has discovered a quality of the atmosphere which is present in all of those atmospheric conditions which our general senses tell us are healthful, and which is deficient in all those conditions which our general senses tell us are not healthful, or which are poorly ventilated.

So it seems very probable, that this ionized condition which correlates with good conditions of ventilation will be proven to have a very vital effect on health and comfort, and that it is a necessary factor in ventilation. We can look forward with high hopes that ionization of the atmosphere will be proven to be the vital characteristic which many have been searching for in the past. Something which in the atmosphere will have the same relation to health and comfort that vitamins have had in food.

DEAN F. PAUL ANDERSON: This paper to me has a very great significance. The author of the paper is one of the young men who started at the Laboratory a number of years ago investigating problems pertaining to ventilation, not in this particular line but the whole subject of what constitutes comfort. He has been a real contributor to the science of heating and ventilating and air conditioning. He is with a great institution now, the School of Public Health which, as you know, is part of the Medical School of Harvard University, one of the great centers of research in this country.

The work that this paper presents is a beautiful tie-in with the things of the past. We didn't imagine when Nicola Tesla years ago presented his thoughts of high frequency and high voltage that some day the heating and ventilating engineer could use this material. The ionization of air is accomplished, as you probably learned from the paper, through the means of producing high voltage and high frequency, not through ionization, which comes in another scale of our energy. After all, this is a great tie-in with the whole theory of energy, the cosmic rays of Millikan, and all of the work on radiant energy that has been done in recent years, finds now to the heating and ventilating engineer a specific application.

There has been a good deal of controversy about this question of mechanical ventilation, what it does to air. You have heard time and time again expressions like this that bringing air through ducts does something to the air. Well, it doesn't do anything to the air to any material extent, but the ions of the air, the carriers of the electric energy, positive and negative, do change when they come into a room, like this. Why the ions in this room are practically eaten up, absorbed by the physical processes of the body in a very short time.

But we can put those ions back in the simplest sort of way by apparatus that is well known to every engineer and so we are facing this very fascinating sort of thing at the present time.

I often use this illustration. We sit at a game of bridge or poker or some other diversion indoors all evening. As our guests depart, we go out on the veranda and get the air. There is something in that air that the ventilating engineer has missed getting, but he can get it and in my opinion this question of putting the vital thing back into the air by some process is simply another addition to this great work that the Society has been doing for all these years.

HENRY TORRANCE: I have a theory that this outside air has a vital effect on the metabolism of the human system. This is based on my own theory and observation and depends largely on the temperature of the air. I don't know why but it has that effect. If a person is outside in the open there is more consumption of oxygen and formation of carbonic acid than when that same person is in the house and I also know that the temperature has a connection with it. I am very glad to have more light on that subject. I wish to say that if that metabolism, or whatever it is, is excessive, it does a human system harm instead of good, because it may chew up

the tissues faster than the digestive processes can replace them. So that as people get old, they may be injured by too much fresh air. That is another theory of mine.

A. R. STEVENSON: I understand that in Germany there have been some experiments made along these lines and that their investigators have reported that the light ions didn't have any special effect, but the heavy ions were the ones they thought had some effect on the heat. I would like to ask Mr. Yaglou if he has done anything along that line.

PROF. YAGLOU: Prof. Dessauer of the University of Frankfort carried out important experiments during the past 10 years, but his work was almost entirely confined to large ions. According to our work, there is a special group of ions, applying particularly to ventilation work. The mobility of this group runs from the maximum possible value in air down to two-tenths of a centimeter per second per volt per centimeter. Dessauer's ions range chiefly from a mobility of seven-thousandths (.007) of a centimeter per second down to eighteen ten thousandths (.0018).

We did considerable work on the physiological and subjective effect of small and large ions, which confirms, more or less, Dessauer's findings. According to our data, the small ions seem to be more important than the large ones, perhaps because their mobility is so high that they readily diffuse to the mucous membrane of the respiratory tract and lungs. However, we are not yet ready to draw conclusions.

J. J. AEBERLY: Many of the members are inclined to look upon this subject as an abstract activity of the Society but this is not entirely so. The ionic content of the air may have a practical significance. Experience in our every-day life, particularly in its relation to health, has set out problems regarding the electrical property of the air which require a solution, and for this reason the Society should not consider these lightly, but on the contrary it should encourage a continuation of this work regardless of whether the results are positive or negative.

Two significant examples in connection with health may be cited to show why scientific men frequently look to the electrical properties of the air for an answer to their perplexities. The medical profession is aware that hydrogen ion concentrations in the blood must be kept within certain comparative narrow limits or sickness will result, and if this electrical property of the blood stream is not again brought back to these narrow limits of hydrogen ion concentrations, death ensues. Again, in the medical profession it is known that the incidence of respiratory diseases is high in the printing and laundry industries. In our research for the cause of this high incidence, particularly in the printing industry, some investigators believe they have found a probable answer in the fact that the printing industry attempts to affect the electrical property of the air by maintaining high humidities to overcome what it calls static and thereby produces a health hazard.

It may be that ultimately, as the subject matter of this paper is more completely investigated, the many complex problems regarding our response to our environmental condition will be answered. There is sufficient doubt regarding certain effects of air conditions, particularly in industrial activities, to justify action on the part of this Society to encourage Professor Yaglou in this splendid work and to contribute liberally to the expense of such further investigation.

MR. COLE: If it is found eventually that it is desirable to ionize air artificially, can ionization be accomplished by the air supplied or might it have to be done by some separate means?

PROF. YAGLOU: The ionic content of indoor air could be brought up to the prevailing outdoor concentration by control of the air supply, but the amount of outdoor air to be furnished (160 cfm per person) is so large as to make this method impractical and expensive. It would be much less costly to make use of a special

apparatus for controlling the ionic content of occupied rooms at any desired concentration, regardless of air supply which in itself is often too low in ion content.

PRESIDENT CARRIER: I believe this paper marks a possible milestone on the pathway of progress in the art of ventilation. It may be the real secret of effective ventilation in which we may reproduce the best climatic conditions.

No. 922

ACOUSTICAL PROBLEMS IN THE HEATING AND VENTILATING OF BUILDINGS

By VERN O. KNUDSEN,[1] PH.D., LOS ANGELES, CALIF.

NON-MEMBER

IT IS not improbable that the physical constituents of our next era of prosperity will include (1) developments in the air conditioning and the mechanical refrigeration of buildings, and (2) the abatement of noise in buildings and out of doors. The problem of noise abatement is destined to become more and more closely associated with the problem of the air conditioning of buildings.

The public has become noise conscious, and is directing an effective attack upon all unnecessary noise. Thus, New York City has an active Noise Abatement Commission which has already made an extensive survey of noise in the streets and in the buildings of New York City. Campaigns are being instituted against the worst noise nuisances such as motor trucks, street railways and automobile horns. Boston and Chicago also are organizing campaigns against noise, and other cities are planning to take steps in the same direction. Manufacturers and distributors of acoustical materials are installing sound-absorptive tiles, felts and plasters in all types of commercial buildings for the purpose of reducing noise. Investigations and researches by efficiency experts of leading corporations, by health authorities, and by life insurance agencies are demonstrating that noise is exacting a heavy toll in the reduced output of all types of workers, in worn and shattered nerves, and in a shortening of the average span of life. The reduction and suppression of noise is a necessary antidote for the machine age in which we are living.

This campaign against noise is of considerable moment to the heating, ventilating and refrigerating industries. In the first place, the equipment used for heating, ventilating and refrigerating is not free from offense as a source of noise; and in the second place, the elimination of noise in many buildings will require that the buildings be air conditioned in order to prevent the transmission of noise through open windows, which at present is the most troublesome source of noise in buildings. It is obvious therefore that the heating and

[1] Associate Professor of Physics, University of California at Los Angeles, Los Angeles, Calif.

Presented at the 38th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Cleveland, Ohio, January, 1932.

ventilating engineer and the acoustical engineer will have many overlapping problems during the next two or three decades. It is therefore very much in the interest of the heating and ventilating engineer to gain a working knowl-

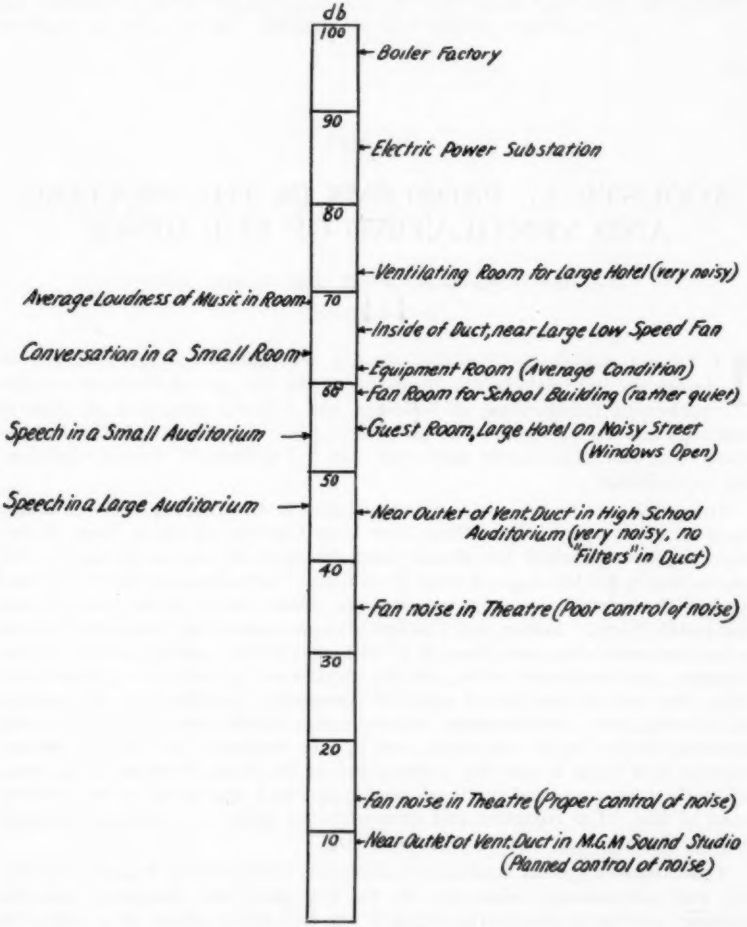


FIG. 1. CHART SHOWING THE AVERAGE LOUDNESS OF SPEECH, MUSIC, AND A NUMBER OF TYPICAL NOISES. NOTE THAT MANY NOISES INCIDENT TO THE VENTILATING OF BUILDINGS ARE CHARTED

edge of the basic facts of architectural acoustics. It is the purpose of the present paper to set forth certain of these facts which should be serviceable to all heating and ventilating engineers.

REQUIREMENTS FOR GOOD HEARING IN INTERIORS

The requirements for good hearing in any architectural interior are the following:

1. The sound, whether speech or music, should have an adequate loudness in all parts of the room.

2. The sound energy should be distributed uniformly in all parts of the room, and the sound reaching the listeners should be free from long-delayed reflections which produce interferences or echoes.

3. The room should contain sound-absorptive materials in such amounts, and of such qualities, as will provide a proper balance between the persistence and the cessation of the articulated components of sound. In other words, the reverberation in the room should be long enough to sustain harmony and impart tonal blending to music, and at the same time it must be short enough to prevent the overlapping and the confusing of the separate sounds of articulated speech.

4. The room should be free from all sources of noise, whether of inside or outside origin.

The amount of sound energy generated by the average speaker is extraordinarily minute. In conversation a person generates on the average only 10 microwatts of speech energy. The average amount of speech energy generated by a speaker in an auditorium is somewhat greater than this, varying from about 25 microwatts in small rooms to about 100 microwatts in very large auditoriums. The acoustical power of unamplified speech is so minute that it would require all of the adult women in the United States speaking *fortissimo* at the same time to generate a single horse power of speech energy. It is obvious therefore that when the speech energy from a single speaker is diffused throughout a large auditorium the amount of speech energy reaching a listener in the auditorium will be almost infinitestimally small. If it were possible to set up a pressure gage in an auditorium which would be sensitive enough to measure the pressure variations in the air owing to the vibrations of sound, such as are produced by speech, the root mean square pressure variation would be less than one millionth of atmospheric pressure. But even such feeble vibrations can be recognized by the sensitive mechanism of hearing provided the acoustical conditions in the room have been made as nearly ideal as possible.

In order to gain a better notion of the loudness of speech and music and of the loudness of noise it is necessary to define a suitable unit for measuring the intensity or loudness of sound. It is becoming more and more universal to rate the intensity of sound in terms of what is called the *sensation level.* Often the term *intensity level* is used to convey the same meaning. The sensation level of a sound is defined in terms of the sound which is just barely audible to a person with normal hearing. Thus, suppose that I_1 be the average intensity of a certain sound in a room. (The intensity is rated in terms of the number of microwatts of acoustical energy fluxing through an area of one square centimeter.) Now suppose I_0 be the intensity of the same sound when it has been reduced to the point that it is just barely audible by the average person with normal hearing. The *sensation level* of the sound having an intensity I_1 is then equal to

$$\text{Sensation level} = 10 \log_{10} \frac{I_1}{I_0} \quad (1)$$

If I_1 be of such a magnitude that it is necessary to reduce its intensity one million fold in order to reduce it to the threshold of audibility, its sensation level will be equal to 60 units. This unit is called the decibel (abbreviated db). The chart shown in Fig. 1 gives the sensation levels of a number of familiar sounds, including several noises which concern the heating and ventilating engineer.

The average sensation level of speech in a small room is about 60 to 65 db,

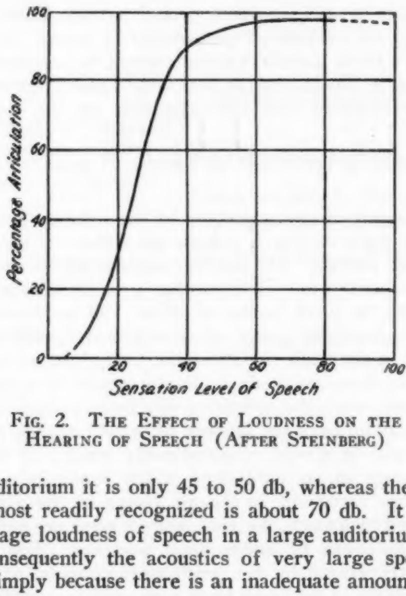


FIG. 2. THE EFFECT OF LOUDNESS ON THE HEARING OF SPEECH (AFTER STEINBERG)

and in a large auditorium it is only 45 to 50 db, whereas the sensation level at which speech is most readily recognized is about 70 db. It is apparent therefore that the average loudness of speech in a large auditorium is at a critically low level, and consequently the acoustics of very large speech halls may be intolerably poor simply because there is an inadequate amount of speech energy for distinct hearing.

The manner in which the recognition of speech depends upon the loudness or the sensation level of speech is shown in Fig. 2.[2] The curve in Fig. 2 is based upon speech articulation tests conducted at the Bell Telephone Laboratories. Meaningless speech syllables are called out into the microphone of a distortionless speech amplifier, and observers listening with telephone receivers connected in the output of the amplifier write down what they hear. The recorded lists are then compared with the called lists. The percentage articulation represents the percentage of the called syllables which are heard correctly. The sensation level of the speech is adjusted by means of an electrical attenuator so that the sensation level of the speech can be maintained at any level between inaudibility and 90 db. It will be noted that at a level of about 45 db

[2] John C. Steinberg, Effects of Distortion Upon the Recognition of Speech Sounds, *Journal Acoustical Soc. of America*, 1, 121 (Oct., 1929).

the articulation begins to drop off very rapidly with any further diminution of the intensity. Since 45 db is about the average level of speech in a very large auditorium it is obvious that any slight downward modulation of the speaker's voice or any slight interference from noise or any other source would interfere very seriously with the hearing of speech. The advantage gained by amplifying speech to an adequate level in large auditoriums is amply demonstrated in the modern cinema theatre. Thus, one experiences practically no difficulty in hearing reproduced speech in the modern "talkie," whereas one often experiences difficulty in hearing all of the spoken lines in the legitimate theatre. The difference is almost wholly attributable to the difference in the loudness level of the speech.

The Effect of Form Upon the Acoustics of a Room

The acoustics of every room is very much dependent upon the shape of the room. Thus, domed or cylindrical ceilings, or concave walls, are likely to give rise to sound foci, and thus the sound will be non-uniformly distributed throughout the room. Again, high ceilings or other large extensions in the dimensions of a room may give rise to delayed reflections. If a reflection be delayed as much as 55 ft behind the direct sound, the reflected sound will interfere with the direct sound, and if the delay be as much as 65 to 70 ft, the reflected sound will be heard as an echo of the direct sound. It is not within the scope of this paper to enter into a discussion of the effect of form upon the acoustics of a room, as the subject is adequately treated in any standard book on architectural acoustics.

Effect of Reverberation Upon the Hearing of Speech in a Room

Reverberation is the prolongation of sound in a room owing to the successive reflections from the boundaries of the room. Thus, in a large room bounded by highly reflective surfaces such as cement or hard plaster the reverberation of an ordinary sound may remain audible 10 to 15 seconds. It is obvious that speech would not be intelligible in such a room since the separate components of speech would overlap and confuse. If the walls of this same room be covered with a highly absorptive material, as an acoustical tile or felt, the reverberation of the same sound would be audible for only a fraction of a second. The time of reverberation in a room is defined as the time required for a sound of a certain pitch to die away to one millionth of its initial intensity. That is, if a tone in a room has an initial intensity of 60 db, then the time required for that tone to die away to inaudibility after the tone has been stopped is a measure of the time of reverberation for that tone. In general, the reverberation time of a room depends upon the pitch of a tone, and is longer for tones of low pitch than it is for tones of high pitch. The time of reverberation can be calculated by means of the formula

$$t = \frac{0.05\ V}{-S \log_e (1 - a)}, \tag{2}$$

where

t represents the time of reverberation,

V represents the volume of the room in cubic feet,

S represents the total interior surface of the boundaries of the room in square feet,

a represents the average coefficient of sound-absorption of the interior boundaries of the room.

The values of a for standard building materials and other acoustical materials can be found in any standard reference book on architectural acoustics.

In Fig. 3 are given the results of speech articulation tests conducted in a number of large auditoriums, of about the same shape and size, but having different times of reverberation. The reverberation times here given are for a tone of 512 cycles per second. It is customary to regard the reverberation

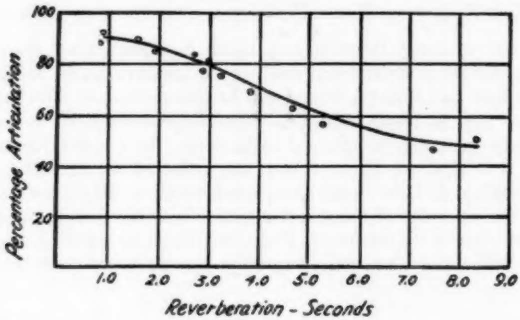


FIG. 3. THE EFFECT OF REVERBERATION UPON THE HEARING OF SPEECH IN LARGE AUDITORIUMS

time of a room as the reverberation time for a tone of this pitch. Thus, the most reverberant auditorium tested in this series had a reverberation time of about 7.5 seconds, whereas the least reverberant auditorium (a modern sound stage for making talking pictures) had a reverberation time of less than one second. It will be noted that the percentage articulation improves as the reverberation time decreases. In fact, the optimal reverberation time for the hearing of speech is approximately one second. For purposes of design, the curves given in Fig. 4 are useful. They give the optimal times of reverberation for tones of different pitch in auditoriums which are to be used for both speech and music. In general, music rooms should have slightly longer, and speech rooms slightly shorter, times of reverberation than the times given by the curves in Fig. 4.

EFFECT OF NOISE UPON HEARING OF SPEECH IN AN AUDITORIUM

The acoustical problem which most concerns the heating and ventilating engineer is the effect of noise upon the acoustics of a room. Experience shows that even very feeble noises interfere with the hearing of speech, and therefore it is necessary that every precaution be taken to eliminate all possible sources of noise in auditoriums. The effect of noise upon the hearing of speech in an auditorium is shown in Fig. 5. Thus, a noise of 50 db will reduce speech

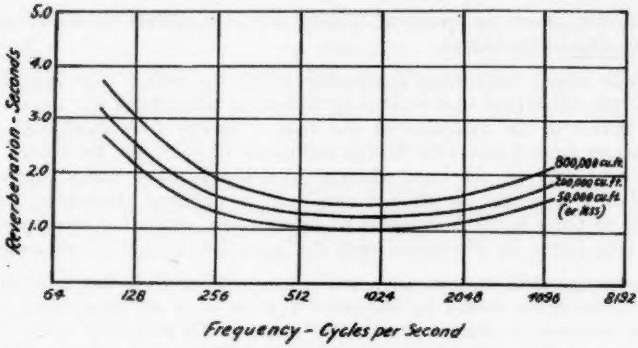


FIG. 4. OPTIMAL REVERBERATION CHARACTERISTICS FOR ROOMS USED FOR BOTH SPEECH AND MUSIC

articulation to approximately 50 per cent, and even relatively feeble noises will produce an appreciable interference. The tests from which these data were obtained were conducted in such a manner that the speaker did not hear the noise. If the speaker as well as the listener hears the noise, the speaker will attempt to raise his voice above the level of the noise, and consequently noise in an auditorium does not produce so great an interference with the hearing of speech as is indicated by the curve in Fig. 5. If there must be some noise in an auditorium it is much better to have this noise in the proximity of the speaker than in the proximity of the listeners, since the noise will then tend to increase the loudness of the speaker's voice, and will not be so bothersome to the audience. However, noise in any form or in any part of the auditorium constitutes a

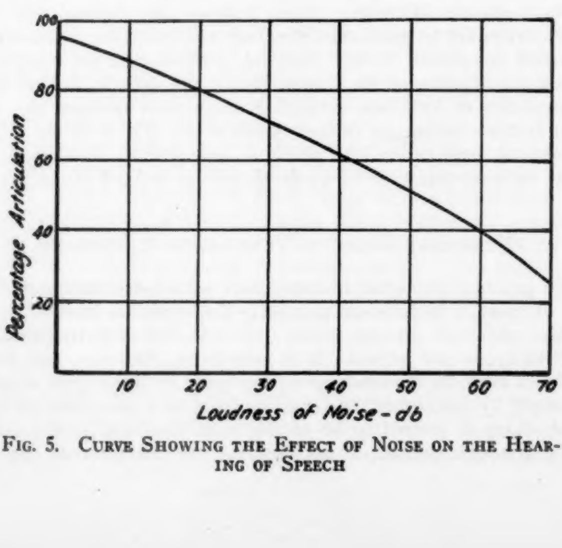


FIG. 5. CURVE SHOWING THE EFFECT OF NOISE ON THE HEARING OF SPEECH

serious impairment of acoustical quality, and every effort should be made to reduce noise to the utmost.

In one school auditorium concerning which the author was recently consulted, the noise from the ventilating equipment constituted the most troublesome defect in the acoustics of the room. Speech tests conducted in this auditorium showed that with the fan and motor in operation, the speech articulation was only 66 per cent, whereas with the fan and motor stopped, the articulation increased to 80 per cent. It is apparent, therefore, that the control of noise in the installation of heating and ventilating equipment is an important factor in connection with the acoustical design of an auditorium.

In every problem of noise control it is necessary to know just how much the level of the noise should be reduced. The amounts of noise which will be readily tolerated in different rooms are given in the following table:

	db
Studios for the recording of sound, as talking picture studios	6 to 8
Radio broadcasting studios	8 to 10
Hospitals	8 to 12
Music studios	10 to 15
Apartments, hotels and homes	10 to 20
Theatres, churches, auditoriums, classrooms and libraries	12 to 24
Talking picture theatres	15 to 25
Private offices	20 to 30
Public offices, banking rooms, et cetera	25 to 40

The values given in this table represent what might be termed ideal conditions, and it is not often that they are realized in existing buildings. However, they represent conditions which can be attained by means of proper design and control of noise, and they represent the limits of noise which should not be exceeded by the noise from heating and ventilating equipment in different types of buildings. Thus, suppose the heating and ventilating contractor is required to guarantee that the noise reaching a theatre from his equipment will not exceed 15 db. Suppose, further, that the contractor knows that the amount of noise at the source end of the duct is, say 60 db. It will then be necessary to introduce acoustical attenuation between the source end of the duct and the auditorium in the amount of 60—15, or 45 db. His problem then becomes an engineering one which is amenable to calculation. Practical methods of meeting this type of problem will be considered in the following sections.

Measuring Noise in Ventilating Equipment

For most practical problems in connection with noise measurement and the insulation of noise, it is sufficient to specify the sensation level of the noise for low, medium and high pitched sounds; for example, for frequencies of 128, 512 and 2048 cycles per second. It is convenient, however, and attempts are often made, to evaluate the loudness of the noise in terms of a single number, as for example by specifying the sensation level of a pure tune of 1000 cycles per second which is judged to be of the same loudness as the noise. Such evaluations, although useful, do not describe the character of the noise, and

if used at all in problems of noise insulation they should be used only with a knowledge of how the noise is distributed throughout the frequency range. Fortunately, most noises have their maximal sensation level within the octave between 512 and 1024 cycles per second, so that if the sensation level of the noise be determined in this range one will have a measure of the noise which is sufficiently accurate for most engineering purposes.

Several practical methods have been developed for the measurement of noise.[3] At a recent symposium on noise measurement conducted by the *American Institute of Electrical Engineers* in April, 1931, eight different instruments for the measurement of noise were presented. Three of these methods will be considered briefly in this section: (1) the tuning fork method, (2) the audiometric method, and (3) the acoustimeter method. The tuning fork method, described by Davis,[4] is extremely simple, but gives quite satisfactory results if used with proper care. Davis used a single fork having a frequency of 640 cycles, but the same type of measurements can be made with several forks. In general, it is advisable to use at least three forks—tuned to say 128 512 and 2048 cycles. The 2048 fork, at least, should be of the Duratone type, which has a very low damping and therefore dies away much more slowly than does a steel fork. Steel forks are quite satisfactory at 128 and 512 cycles. First of all, the forks must be calibrated. It is necessary to know the intensity, in db, of each fork immediately after it has received a standard blow or excitation. If the fork be allowed to fall from a vertical position through an arc of 90 deg, hitting a suitable pad (such as soft rubber or felt for the low pitched forks and hard rubber for the high pitched forks), with a little practice the initial intensity can be reproduced to an accuracy of 1 or 2 db. It is then necessary to know the rate of decay of the forks. In general, the rate of decay in sensation units will be a constant number of decibels per second. The rate of decay can be determined very readily in any well-equipped acoustical laboratory. The 512 steel fork will have an initial intensity of about 70 or 80 db, and it will decay at a rate of about 1.2 to 1.7 db per second. Lower pitched forks decay more slowly and higher pitched forks decay more rapidly than does the 512 fork. In a perfectly quiet room steel forks may remain audible, when held close to the ear with the broad side of the prong toward the opening of the ear canal, from about 50 to 70 seconds, and Duratone forks will remain audible a hundred seconds or longer. If the rate of decay in db per second and the duration of audibility in a *quiet* room have been determined for a fork, its initial intensity will be equal to the rate of decay times the duration of audibility. Thus, if the rate of decay for a fork is found to be 1.1 db per second, and the duration of audibility after it has been given a standard hit is 60 seconds, the initial sensation level of the fork will be 66 db above the threshold of the individual making the measurements. The hearing acuity of this individual should then be compared with the normal (by means of a calibrated audiometer) and an appropriate correction applied to the initial intensity level of the fork.

[3] E. E. Free, Practical Methods of Noise Measurement, *Journal Acoustical Soc. of America*, 2, 18 (1930). R. H. Galt, Results of Noise Surveys—Noise Out of Doors, *Journal Acoustical Soc. of America*, 2, 30 (1930). R. S. Tucker, Noise in Buildings, *Journal Acoustical Soc. of America*, 2, 59 (1930). J. S. Parkinson, Vehicle Noise and Noise Reduction, *Journal Acoustical Soc. of America*, 2, 65 (July, 1930).

[4] A. H. Davis, Measurement of Noise by Means of a Tuning Fork, *Nature*, 125, 48 (Jan. 11, 1930).

How to Measure Noise

The method of measuring any noise is as follows: The observer, in the presence of the noise, strikes the fork a standard blow and at the same instant starts a stop watch. The fork is then held in front of the ear canal, moving it back and forth slightly, until the tone of the fork is just completely masked by the noise, at which instant the watch is stopped. This measurement is then repeated two or three times and the average of all readings will make it possible to calculate the sensation level of the noise at that particular frequency.

Thus, suppose that the fork has an initial intensity of 66 db, and that its tone decays at a rate of 1.1 db per second. Then if the fork remains audible only 20 seconds in the presence of the noise, it means that the masking effect of the noise at this frequency is 66—22, or 44 db. Similar readings are obtained at other frequencies, and the resulting data will give an approximate audiogram of the measured noise. Measurements of this type, made with 128, 512 and 2048 forks, give a very satisfactory description of the intensity and frequency distribution of different types of noise. Both the apparatus and the method of measurement are extremely simple and are fairly reliable.

A somewhat similar and also a very convenient method for making an approximate measurement of noise is made possible by means of a buzzer type of audiometer, such as the Western Electric 3A Audiometer. The buzzer in this audiometer has a fundamental frequency of 160 cycles, and it has an abundant supply of overtones, so that its noise is quite representative of most noises produced by heating and ventilating equipment. The receiver of the audiometer is equipped with an off-set receiver cap so that the ear of the observer hears both the noise of the audiometer and the noise which is to be measured. The observer first determines the amount of attenuation (in db) which must be introduced in the audiometer to reduce the buzzer sound to inaudibility when listening in a perfectly quiet place. He then makes a similar measurement when listening in the presence of the noise which is to be measured. The difference in the two readings of the audiometer, in the quiet place and in the noise, then gives a rough measure of the masking effect of the noise. The masking effect of the noise, as measured by the threshold shift on the audiometer dial, will usually be about 5 to 10 db less than the sensation level of the noise.

In the acoustimeter method of measuring noise, the noise vibrations are picked up by a microphone, amplified by a vacuum tube amplifier, passed through a frequency weighting network, and registered or recorded by a suitable galvanometer. The frequency weighting network is based upon the sensitivity and sensibility characteristics of the normal human ear, so that the deflections registered by the instrument will be comparable to those which would be heard by the ear. A single reading of the instrument will therefore give a fairly reliable measure of the "noisiness" of the noise, although the effect of the frequency weighting network will introduce certain errors owing to variations of intensity and frequency distribution in different noises. Such an instrument is especially useful for obtaining a continuous record over an extended period of time.

PRACTICAL METHODS OF REDUCING NOISE IN HEATING AND VENTILATING EQUIPMENT

In a recent paper the author discussed practical methods of controlling noise in heating and ventilating equipment.[5] The specific noise problems which confront the heating and ventilating engineer are the following:

1. Improvements in the quiet operation of equipment.
2. The location and insulation of the equipment room so that no noises are transmitted through the walls, the ceiling, or through openings, into rooms where quiet is required.

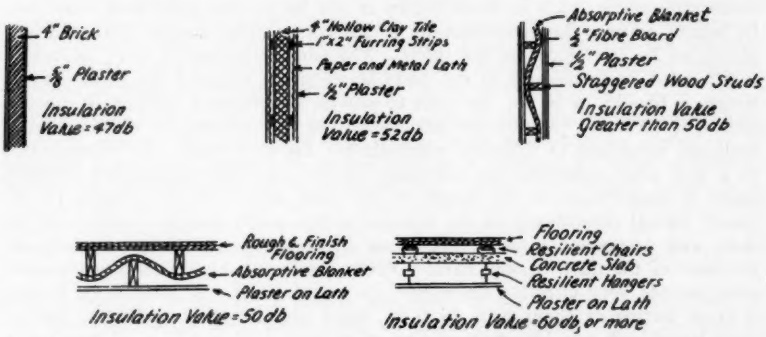


FIG. 6. THREE WALL SECTIONS AND TWO FLOOR AND CEILING SECTIONS WHICH WILL PROVIDE A SOUND-INSULATION OF MORE THAN 45 db.

3. The isolation of all vibrating or rotating equipment, so that the vibrations will not be communicated to the solid frame of the building.
4. The prevention of noise transmission through the ventilating ducts.

In order to illustrate the manner in which the noise incident to a typical installation of air conditioning equipment can be controlled, suppose that it is required to provide a quietly operating system of ventilation for a theatre. Preferably, the equipment room should be located at a considerable distance from the theatre auditorium, but this is not necessary if proper steps be taken to insulate the noise which may be transmitted through the walls of the equipment room and also the noise which may be transmitted through the ducts. All rotating or vibrating equipment should be mounted on flexible supports which are so resilient that the natural frequency of the mass of the equipment on the flexible support will be low in comparison with the frequencies of the vibrations generated by the equipment. If the fundamental frequency of vibration produced by the machine be 100 cycles per second, the natural frequency of the mass of the machine on its flexible support should not be greater than about 20 cycles per second. Since the noise level in the theatre should not exceed 15 db, it is necessary to supply sufficient insulation to reduce the noise

[5] V. O. Knudsen, How Sound Is Controlled, *Heating, Piping and Air Conditioning*, 3, 10, 815-820 (Oct., 1931).

to this level. Suppose that measurements have been made of the noise generated by the ventilating equipment in a typical equipment room and that these measurements reveal a noise level of 60 db in the equipment room and a noise level of 65 db in the duct at its input end. If the equipment room is adjacent to the theatre auditorium it will then be necessary to design the walls and ceilings of the room in such a manner as will provide an effective insulation of at least 45 db between the equipment room and the auditorium. In Fig. 6 are shown three types of wall sections and two types of floor and ceiling sections which will supply an insulation in excess of 45 db. With properly designed walls and ceiling, such as those shown in the figure, the equipment room can be located as near the auditorium as is desired, provided further that sufficient insulation be introduced in the ducts. The sound attenuation (or reduction) which must be introduced in the ducts between the fan and the theatre auditorium is 65—15, or 50 db. In order to provide the required amount of attenuation, it is necessary to know the attenuation per foot of ducts of different cross sectional areas and of different materials for the duct walls. In general, and to a first approximation, the reduction (in db) of noise transmitted through ducts is proportional to the length of the duct, inversely proportional to the "size" (lineal dimensions, as the average of the width and the height) of the duct, and proportional to the coefficient of sound-absorption of the interior surfaces of the duct. Long, narrow ducts, lined with highly absorptive material, provide very effective *attenuators* or *sound filters*. A small duct, having a cross section of about 4 in. by 6 in., lined with a highly absorptive felt or fibre board, will provide an insulation of at least 1 db per foot. Quantitative information is needed concerning the amounts of attenuation or noise reduction provided by different types and sizes of ducts. Manufacturers of ventilating equipment should obtain data of this sort, as these data would make it possible for engineers to provide a specified amount of noise reduction in any duct system, and thus in every installation it would be possible to design in advance of the construction the type and length of ducts required to effect a proper reduction of the noise. Already some of the leading makers of air conditioning equipment have procured the necessary data for coping with the problem, and are routinely installing in the ducts *noise filters* which are designed to meet the noise requirements for each installation. Some useful data and suggestions will be found in a recent paper by Larson and Norris.[6]

DISCUSSION

PRESIDENT CARRIER: I think that this phase of sound prevention, sound absorption, is a most important one in our profession. All ventilating work, especially where we are using refrigeration, requires various periods of vibration from the very high frequency that we get, for example, in the rush of air through ducts and spray nozzles, down to the low frequency of a slowly-driven refrigeration compressor. We have a great range of problems, probably one of the most mysterious being the absorption of sound from fans. As engineers we know little about this subject so we hope that those who are interested in the solution of these problems will ask questions. We will have the written discussions first.

[6] G. L. Larson and R. F. Norris, Some Studies on the Absorption of Noise in Ventilating Ducts, A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931.

J. S. Parkinson[7] (Written): Dr. Knudsen has laid down the requirements for noise control and has indicated the desirable limits for extraneous noise in various types of rooms, also has shown how careful planning may serve to satisfy requirements. Many problems occur, however, in existing buildings, where the remedy is not at first obvious. It may be of interest to detail briefly the methods which the engineer uses in diagnosing such problems and remedying the condition.

The first step, of course, is to determine the seriousness of the problem. Thus, if excessive noise has been reported in the ventilating system of a theater, the engineer first measures the noise level at various locations and compares it with the level at which the sound is being reproduced. This will tell him how much interference is taking place, and how drastic must be his corrective measures. The measurement may be made by any one of the several methods which Dr. Knudsen has outlined. Most common practice today is to use some form of acoustimeter apparatus, properly weighted to make the results correspond to the ear sensitivity.

This survey will probably also tell the investigator the route or routes by which sound is entering the theater. If it is coming from the ventilating grilles, as is probable, it becomes of interest to discover whether it is largely air borne or whether it is being transmitted via the walls of the duct. A device commonly known as a vibration pickup is often used for the latter measurement. Most of these are electrically operated and may be substituted in the acoustimeter circuit for the microphone. Complete information may thus be obtained as to the amplitude of vibration both in the air and in the duct walls.

An inspection of the fan or blower equipment, of the plenum chamber, and of the duct construction will usually yield considerable information. The machinery, as has been said, must be properly insulated against the transmission of vibration. This means some sort of flexible or resilient support. Such a support must of course be carefully laid out so that it is neither over-loaded nor under-loaded. The physical properties of insulating materials are for the most part known, and, where necessary, the engineer should check the layout against equipment live and dead loads. At the same time he must make sure that conduit or pipe connections have not been run directly from the machinery to the building structure.

In a similar fashion all direct structural connection between the fan or blower and the ducts themselves must be eliminated. Here again the vibration pickup or a modification in stethoscope form for use with the ear, is useful in tracing the passage of vibration. For example, a direct connection via conduit is not necessarily a serious fault, unless the conduit is connected to some surface which can carry the vibrations to the duct or which will vibrate over sufficient area to set up air waves.

If there appears to be no structural transmission, the investigator will go over the equipment carefully for excessive vibration which might send sound into the ducts, watching carefully again for any surfaces of sufficient size to serve as sound generators. Motor and fan housings, panel mountings for switches, etc., are frequent offenders. Such surfaces, if found, may be damped by lining with felt or by applying weight at the proper points.

When all such preliminary precautions have been observed, there remain two possibilities—either a more efficient isolating construction may be built under the machines, or else the ducts may be lined with absorbent material to a sufficient distance to eliminate sound transmission. Recent experiments with various methods of point support under machinery have yielded interesting results. For example, in the case of a platform built up on special felt lined supports, it was found that there was a reduction of more than a hundred times in amplitude between the vibration of the platform top and the slab below.

[7] Acoustical Engineer, Johns-Manville Corp.

The most convenient material for duct lining is usually of the felt type, since it can be worked around corners and into special shapes, curves, etc. Such absorbent lining is more effective when placed close to the source of noise, assuming of course that the sound is air borne. It must be remembered that the effectiveness of such an absorbent lining does not increase proportionately with the length of duct treated, the first few feet being the most effective. Where space does not permit sufficient length of treatment, honeycombs may be installed to give greater efficiency.

C. A. ANDREE,[8] PH.D. (WRITTEN): It is difficult to add anything constructive to a paper so thorough and explicit as that which has been presented by Professor Vern O. Knudsen. It occasionally happens, however, that the additional emphasis and slightly different view-point which another may offer will serve to enhance the value of the original paper and may even widen its appeal. With this in mind, the fol-

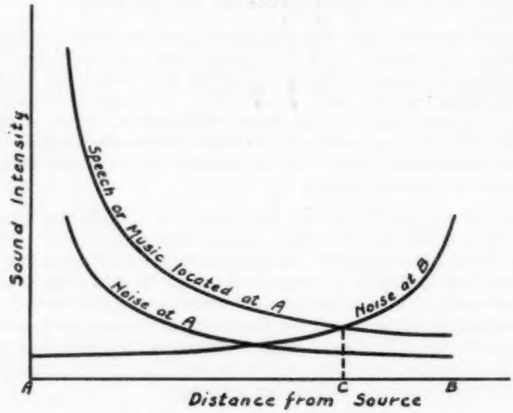


FIG. A. INTENSITY OF SOUND

lowing comments, which appear to emphasize rather than to detract from the original paper, are submitted.

Professor Knudsen's point that a source of noise which is located near the speaker is less bothersome than one which is located near the audience may be worthy of further emphasis.

Fig. A shows how in a properly designed auditorium the intensity of the sound from a source of speech or music located at *A* decreases rapidly until it is a minimum at some remote point *B* in the auditorium. Now the intensity from a source of noise such as a ventilator will decrease in a similar fashion as one moves away from the origin of the noise. If such a noise source be located near *A* and if it does not interfere with the reception of speech or music at *A*, it will not interfere with the reception of speech or music at any point in the auditorium. If on the other hand the same source of noise be located at *B*, there will be a region *C,B* over which the intensity of the noise may exceed that of speech or music and within this region the reception of speech or music will be entirely unsatisfactory. This serves then merely to strengthen the argument which Professor Knudsen makes that a noise source is less objectionable near a speaker than near the listeners.

[8] Dept. of Electrical Engineering, University of Wisconsin.

Any haphazard attempt at noise reduction will very likely end in dissatisfaction and only a very thorough going job is likely to be worth while. To illustrate this point let us suppose that two sources of noise are present in an auditorium. One source may enter the auditorium through the ducts of the ventilating system; a second may arise from a motor, pump, elevator, or may even be the mechanical vibrations of the fan of the ventilating system transmitted through the structure of the building. To make the illustration simple let us suppose that each source of noise contributes the same amount to the total noise in the room and that the total noise level is 50 decibels. Let us suppose that in order to economize it is decided to eliminate the source of noise from the ventilating ducts but to leave the other source of noise untouched. A very thorough piece of work is done on the ventilating ducts and as a result the noise from this source is reduced to zero. Now the total noise remaining in the auditorium has not been cut in half as might be supposed but has only been reduced to 47 decibels.

While this relation may seem a trifle strange to some, it nevertheless is an experience with which we are all quite familiar in other fields. To give just one example, we observe that if a room which we wish to make dark is lighted by two equal lights, we gain but little if but one light is turned out. We must turn out both lights if we wish to obtain any real effect. In the same way we must eliminate *all* appreciable sources of noise before we obtain a decided effect in the elimination of the total noise. It is a curious fact that there is a very close relation between the ordinary wire gage and the decibel scale in acoustics, and this close relation may be of assistance to those who have difficulty in understanding the scale used by acoustical engineers. If we compare the diameter of a wire in a wire gage with the pressure in a sound wave, then the area of the wire will compare with the intensity or energy of the sound wave and the number of the wire in the wire gage will compare with the decibel scale used by acoustical engineers. To make the analogy complete we must renumber the size of the wires. In the decibel scale the zero of the scale represents the smallest sound in which we are interested. In the wire gage we have been less consistent for the zero of the scale represents one of the largest wire sizes in which we are interested. If we renumber our wire sizes and let zero represent wire size No. 40, and if we let No. 40 represent what is normally zero, our analogy will be complete. Thus, if we cut the area of the wire in half, the wire size drops by three. In the same way if the sound energy is cut in half, the decibel scale drops by three.

Problems which await the combined efforts of acoustical and ventilating engineers for their solution are common. Amid all the luxury and refinement of the modern hotel, it is still an almost impossible task to find, near the center of any of our large cities, a quiet, well-ventilated room wherein one may enjoy a quiet night's rest. The solution of this problem is now available in the form of ventilators which filter out the noise and dirt from the outside air and may be readily applied to any window.

The use of desk fans and similar devices as an aid to ventilation is always a source of annoyance because of the accompanying noise.

At present there is a trend away from the use of centralized ventilating equipment and toward the use of equipment which is localized and furnishes ventilation for relatively few rooms. Such equipment furnishes new problems in the elimination of noise.

The solution of some of these problems, as I have indicated, is already at hand, and the solution of many others will surely yield to research and the application of the principles outlined in this paper.

O. B. HANSON [9] (WRITTEN): The experience which I had in the design and construction of studios at 711 Fifth Ave. in 1926 and 1927 demonstrated only too well

[9] Mgr. Plant Operation and Engineering, National Broadcasting Co.

the lack of knowledge on the part of architects and contractors, with respect to sound control. It was perhaps even worse that what little information they did have was based on an erroneous impression and a total lack of understanding of the science of sound. Although fortified with much theory, we had little precedence to go by in the successful application of these principles.

We ran into a similar problem in the design and development of the Chicago plant, and the urgent need of study on the part of architects, building engineers and ventilating engineers, of the science of sound control, is still very apparent.

The application of sound absorbers to ventilating systems to prevent the transmission of fan noises along the ducts, and the prevention of sound transmission from one room to another along ventilating ducts, is becoming very important, if only from the standpoint of reducing noise. In the case of broadcasting studios it is a vital problem, as noise levels in broadcasting studios must be held down to a minimum and the transmission of program material through partitions must be absolutely nil. Likewise, the transmission of program material through air ducts of a ventilating system, from one studio to another, connected to a common ventilating system, must be prepared to have an attenuation of some 80 to 100 decibels. Sound insulated walls, floors and ceilings of broadcasting studios must also have a sound attenuation of something on the order of a hundred decibels.

The sound insulating principles of floating walls explained in Dr. Knudsen's paper are fundamentally those used in broadcasting studios, except that they are applied to heavy masonry walls, as the requirements are far more severe than those of an office or apartment house.

Since the advent of broadcasting and the entrance into the sound field of the motion picture art, the public is becoming more and more sound conscious and are realizing how injurious noise can be to their every-day life.

V. L. CHRISLER [10] (WRITTEN): The control of noise from air conditioning and refrigeration machinery for buildings is of considerable importance as it is believed that many of the better houses will be provided with this kind of equipment in the near future provided such installations can be made to operate quietly.

The demand for such equipment will arise for two reasons. The one appealing most at the present time would be temperature control during the summer especially in the warmer portions of this country. If the first installations are properly made and some changes made in the windows, dirt and street noises can also be largely eliminated. The elimination of noises especially in downtown apartments and hotels, in any place where the noise level is high, will in turn create a demand for more such installations. The success of such an installation, however, depends upon the noise of the machinery being controlled so that very little of it passes through the air ducts to the rooms.

As Dr. Knudsen has pointed out, the noise from the fans and refrigerating machines can be eliminated by the proper lining of the duct, the use of insulating bases, etc. At the present time, however, there is not sufficient general information on this subject. It is felt that a more intensive study of the subject should be made so that satisfactory installations can be made. At the present time many of the installations are very unsatisfactory from the standpoint of the amount of noise produced.

S. K. WOLF [11] (WRITTEN): Dr. Knudsen points out that the current campaign against noise is of much importance to the heating, refrigerating and ventilating industries, as it is to many other industries producing goods for public consumption. Ventilation engineers should not contribute to the common good with one hand, and, unknowingly, take back in the other, a portion of their benefits in the form of

[10] Physicist, Bureau of Standards, U. S. Dept. of Commerce.
[11] Mgr. Acoustic Consulting Dept., Electrical Research Products, Inc.

noise and its attendant ills. The conclusion then is that it would be to their best advantage to possess a usable knowledge of the fundamentals, at least, of acoustics.

Noise measurement is not a simple problem. The novel tuning fork method described here involves a minimum of apparatus, and, at the same time, gives an idea of the frequency distribution of the noise under test. For these reasons the method is highly attractive. At the same time, since it involves masking effects of complex sounds, results may be occasionally misinterpreted by those unfamiliar with such phenomena. For example, higher intensities of low frequency components of about 50 db or more will cause masking at much higher frequencies, apparently indicating the presence of non-existent high frequency components. However, such difficulties are common in acoustical work and a few could be named for most methods of noise measurement. The ideal measuring instrument is yet to be devised.

A figure of 15 db is used in the paper as the desirable noise level in a theater. It should be pointed out that this is not very often realized even in what are commonly considered quiet locations. In a theater, noises from the street, projectors and other machinery usually are at a level of 25 db or greater. This last figure seems a reasonable value to work for as the noise contributed by an audience is more than likely to be in excess of it, except for quieter moments. Incidentally, it may be remarked that to keep the noise level at 15 db it is not necessary that the noise from individual sources be reduced to this amount. The number of sources and the acoustic absorption of the theater surfaces are influencing factors to be considered. However, if the procedure described by Dr. Knudsen is followed and the noise from such sources is not allowed to exceed 15 db, the disturbance in their immediate vicinity will be little greater than at other points in the theater, a highly desirable condition.

C. F. EYRING (WRITTEN): In my opinion, three aspects of this paper should be emphasized.

From a listener's point of view it is better practice to reduce noise than to overcome its masking effect by amplifying speech. Among engineers, therefore, noise reduction should be as popular, if not more popular, than speech amplification. In large auditoriums, the speech power of the average speaker, without the aid of an address system, is not sufficient to bring the sound intensity to the level where good articulation is obtained, even under the ideal condition of no noise. But the use of such a system to raise the voice above the extraneous sounds of a noisy room will not eliminate the annoyance of the noise during the silent intervals even if the articulation is improved by the speech amplification. Silent intervals should be silent and not filled with the distracting effect of noise.

Manufacturers of ventilating equipment would do well to standardize the methods of measuring the noise reduction produced by definite sizes and types of ducts. Only by such a standardization can the purchaser of ventilating equipment be able properly to judge the superiority of various types of equipment.

The science of acoustics has now progressed far enough so that the manufacturer of ventilating equipment may have confidence in the information received from an engineer trained in acoustics. An acoustical engineer, therefore, should find a useful place in an organization built for the solving of heating and ventilating problems.

R. H. GALT[12] (WRITTEN): This comprehensive paper deals with one of the most serious problems of the heating and ventilating of buildings, namely the suppression of noise due to fans, motors and other appliances. It is a common experience today in churches, lodge rooms, public auditoriums, restaurants, and even in theaters, to find the noise from these sources so loud as to interfere seriously with the understanding of the spoken word, with the appreciation of music, and with the dramatic effect of intervals which should be periods of silence.

[12] Bell Telephone Laboratories.

There are two aspects of noise which should be appreciated by heating and ventilating engineers. In its positive aspect, noise indicates the existence of sound waves which should never have been produced, or, if unavoidably produced, should have been suppressed before reaching the room; these sound waves are annoying, distracting, and at times even injurious to auditors, and definitely limit the attainable degree of quietness. In its negative aspect, noise interferes with the reception of all the desired sounds, such as those of music and speech. This action takes place in the ear itself. The noise, while it lasts, renders a person partially deaf, so that sounds below a certain intensity are not heard at all, and the perception of louder sounds is impaired. From both of these standpoints noise is objectionable, hence its suppression is doubly significant.

E. C. LLOYD (WRITTEN): The use of cork as an isolating material under moving machinery is common practice. It is true, as Dr. Knudsen has so well pointed out, that a spring suspension can be made to meet the conditions of vibration imposed by the moving machine. It is equally true that cork pads can be designed to meet all conditions ordinarily met in heating and ventilating practice. It is true that in past some cork pads have been so installed that the vibration has been amplified rather than reduced. It is equally true that with today's knowledge of this subject, cork isolated foundations can be designed to meet all conditions. The case where cork isolators have amplified vibration in past is a rare occasion rather than the rule, as it is well known that thousands of installations are in satisfactory service in all parts of the world.

There is one other point that our present day knowledge on this subject enables us to do and, that is, to design the type of isolation necessary to give the greatest reduction in vibration. In the past any reduction was welcome. Today the maximum reduction can be definitely assured with proper design of isolation.

F. L. HUNT (WRITTEN): The increasing attention which has recently been given to the industrial significance of noise and the importance of noise abatement has stimulated interest in studies of methods of noise measurement. Instruments based on the audiometric method of sound masking at the ear have been available for some time. They are relatively simple in construction and compact and serve admirably in many cases where the highest accuracy is not required. In recent acoustic studies at the Bell Telephone Laboratories increasing use has been made of the acoustimetric method in which the output of a microphone is amplified, with loudness weighting, rectified and then applied to a suitable indicator or recorder. A calibrating circuit is included to check the readings. This method has the advantage of greater accuracy and reproducibility since it is free from the personal equation involved in masking methods. An instrument of this type designed for general sound intensity measurements is now available.

F. R. WATSON [13] (WRITTEN): This paper gives valuable information and should serve to awaken interest in quieting of buildings. There is an increasing objection to noise because of its evil effects on people resulting in nervousness, sleeplessness and lowered efficiency; so that a concerted movement to control sound in buildings becomes more imperative. It is interesting to read proposals to construct buildings entirely of glass and steel. Unless some provision is made to control sound, these structures will be uninhabitable.

Dr. Knudsen presents a systematic account of the noise nuisance, and sets forth practical suggestions for its control. More data are needed on certain types of ventilators, and it is hoped that various companies will be interested to finance investigations to secure this useful information.

L. L. SMITH: I am greatly interested in Dr. Knudsen's paper, especially from the designing standpoint. The charts shown indicate that there are a certain number of

[13] Dept. of Physics, University of Illinois.

decibels of noise generated by equipment. We talk a great deal about insulating machinery to eliminate vibrations to and through the floor, and provide acoustical treatment of rooms for reverberation and noise. I wonder then after we have done this, how many of the decibels generated are coming into the room by the air noise?

Dr. Knudsen mentioned that we can eliminate the noise coming into the room or at least a percentage of it by proper design of the ducts and selection of material. Of course, we have used sheet metal for ducts and we have at times lined them, maybe for a distance of 10 to 20 ft, in an effort to absorb the noise as it left the fan. I wonder if that has been very effective? We have also successfully used sound chambers with certain baffles to eliminate noise. I am wondering too what is the correct design for ducts especially the perimeter and shape which can be accommodated in the buildings of today.

The engineer may start out with the assumption that he would like to keep his velocities low so as to be sure to have air at the end of the duct with the power that he is supplying and also to keep down below what he believes is the velocity of noise limits. The first thing that the engineer encounters is the condition that no space remains in the building to accommodate the duct. He may design it for a certain shape, such that the perimeter represents the minimum amount of friction for the cross sectional area of the duct. Then he discovers that the duct must be flattened or changed to various shapes to get through building construction. In the end, unless it is a perfectly open building where exposed duct work is not objectionable from the appearance standpoint, the question is just how far he can go with proper design of duct work to eliminate the noise, even though he has the complete data at his finger tips. Just what is the proportion of decibels that we can practically take care of in our design?

DR. KNUDSEN: If you specify the noise of a fan at the fan end of the duct, it is possible to design filters which reduce the noise to any required extent. There are several types of filters which can be used. The simplest is the one to which I referred in the paper, in which the attenuation, that is the fractional loss of sound energy per foot, can easily be made as great as one or one and a half db per foot. This is accomplished by using small cellular conduits lined with highly absorptive material. They make a honey comb structure of the duct. They may reduce as much as 10 or 15 per cent the effective area of cross section of the duct. They do therefore require a slightly larger space for the duct. The matter of providing an additional 10 or 15 per cent cross sectional area for the duct for a length of 20 or 30 or 40 ft is perhaps all that is needed.

A paper [14] published in the TRANSACTIONS of the A.S.H.V.E. contained some useful data on the amount of attenuation or loss per foot or per 10 ft length of ducts of different sizes with different materials.

Certainly the thing which your Society needs is more specific information concerning the attenuation per foot for ducts of standard sizes and standard materials. The problem, as President Carrier has stated, is a thing which can be handled in an engineering manner at a small cost; and the amount of noise that gets into the room can be reduced to any specified level. If you tell an engineer how many decibels you want your noise reduced, he should in a few minutes calculate the kind and extent of attenuators required to reduce the noise 10, 20, 30, 40, or more decibels, as is needed. In general the reduction in decibels is approximately proportional to the length of the duct, to the perimeter of the duct, to the coefficient of sound absorption of the material of which the duct is made, and proportional inversely to the area of cross section of the duct.

---

[14] Some Studies on the Absorption of Noise in Ventilating Ducts, G. L. Larson and R. F. Norris, A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931.

That is the basis of your problem from the standpoint of physics, and the rest of it is just a simple matter of getting some engineering data.

A. R. STEVENSON: If you wanted to measure the noise of a particular piece of apparatus, how would you measure it? I have found it is necessary to specify that it makes so many decibels at 6 in. or so many decibels at 2 ft or 3 ft and then I discover that if I measure it in one room that has one kind of absorptive walls, I get one answer, and if I put it in another room that has hard walls, I get still another answer.

DR. KNUDSEN: If you know the total flux of sound energy from any sound source and the boundary materials of the room, you can calculate what the average intensity of sound energy will be in that room. The average intensity is just inversely proportional to the total absorption in the room, so that it is possible to calculate the effects produced by different rooms. If you are close to the source it will be necessary to make a correction for the inverse square law of decay of the sound as you go away from the source. There are, to be sure, very great fluctuations in the room from point to point owing to the interference pattern set up by reflections from the boundaries, but you can get a statistical average which will suffice for practical purposes. For example, the separate readings, say at the input of the duct may be between 60 and 70 db. If you take the average, 65, certainly from all engineering standpoints that is good enough.

No. 923

HEAT TRANSMISSION AS INFLUENCED BY HEAT CAPACITY AND SOLAR RADIATION

By F. C. Houghten [1] (MEMBER), J. L. Blackshaw [2] (MEMBER), E. M. Pugh [3] (NON-MEMBER) and Paul McDermott [4] (NON-MEMBER), Pittsburgh, Pa.

THE fact that solar radiation is an important factor in the mechanism of heat flow into and out of buildings has long been recognized by heating and ventilating engineers. It is usual, however, in practical heat transmission problems to neglect the sun effect entirely, to employ coefficients determined under steady state conditions, and to consider the average inside and outside temperatures constant throughout the day. With this practice, factors are used to take care of wind, sun effect or its absence, or other unusual conditions in order to insure the selection of a heating plant sufficiently large to handle the load on the coldest day, or an air-conditioning system of sufficient capacity for the warmest weather.

Research, carried on by the Research Laboratory of the American Society of Heating and Ventilating Engineers at the Pittsburgh Station of the United States Bureau of Mines, has shown that a large error [5] may be introduced into the calculations by failure to consider the periodic character of heat flow resulting from the diurnal movement of the sun and the heat capacity of the structure, which determine the timing and magnitude of the heat wave flowing through the walls into a building on a hot, sunny day.

Scope of the Present Work

This paper reports the characteristics of heat flow through 8 roof panels which were studied at the Laboratory during the summer months of 1930 and 1931. It gives the cyclic heat flow through the inner surfaces of these panels, the maximum and minimum rates of penetration during 24-hr periods,

[1] Director, Research Laboratory, American Society of Heating and Ventilating Engineers, Pittsburgh, Pa.
[2] Research Engineer, Research Laboratory, American Society of Heating and Ventilating Engineers, Pittsburgh, Pa.
[3] Associate Professor of Physics, Carnegie Institute of Technology, Pittsburgh, Pa.
[4] Research Engineer, Research Laboratory, American Society of Heating and Ventilating Engineers, Pittsburgh, Pa.
[5] See Bibliography, a.
Presented at the 38th Annual Meeting of the American Society of Heating and Ventilating Engineers, Cleveland, Ohio, January, 1932, by F. C. Houghten.

the timing and intensity of the temperature waves at the outer and inner surfaces, and the rate of heat flow into an air-conditioned space, as these are determined by the thickness, conductivity and heat capacity of the different panels studied.

A theoretical mathematical analysis of the cyclic flow of heat under the foregoing conditions results in the presentation of formulae which give the rate of heat flow into an air-conditioned space for any hour of the day as a function of a Fourier series analysis of the outside surface temperature, the air-conditioned temperature, and the thickness and the physical properties of the structure. An empirical method of determining certain constants in these theoretical formulae is developed. This simplified method gives the heat flow into an air-conditioned space with a fair degree of accuracy in terms of the maximum and minimum outside surface temperatures for the day, the inside air-conditioned temperature, and the thickness and physical properties of the structure.

The paper also includes data on the conductivities, densities and specific heats of the materials contained in the different roof panels and the methods developed

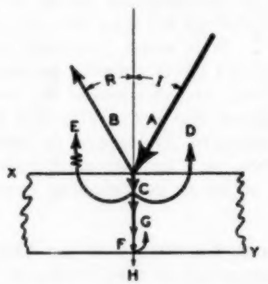


FIG. 1. FACTORS AFFECTING HEAT FLOW THROUGH A ROOF

for obtaining them. A recording pyrheliometer, developed at the A. S. H. V. E. Laboratory for measuring solar radiation, is described, and data are presented for the first days of July, August and September, on the intensity of solar radiation in Pittsburgh, and of the resulting intensity of solar radiation impingement upon a horizontal surface, and upon east, south and west walls.

FACTORS AFFECTING PERIODIC HEAT FLOW

The flow of heat into an air-conditioned space through a roof or wall on a hot summer day, because of the higher outside air temperature and the solar radiation on the outer surface, is very complex. It may be illustrated in Fig. 1, where *XY* is the roof or wall structure. The impingement of solar radiation against the outside surface is represented by *A*. If it were not for the earth's atmosphere with its clouds, haze and dust, the intensity of the solar radiation in Btu per square foot per hour perpendicular to the direction of the sun's rays would be practically constant, showing only a small seasonal variation caused by the change in the sun's distance from the earth. The intensity of solar radiation is decreased by absorption by the earth's atmosphere, especially

in the morning and evening, when the sun must shine through a much greater thickness of the atmosphere in reaching the earth's surface. The Weather Bureau has reported 338 Btu per hour per square foot normal to the direction of the sun's rays for a clear day at Washington. The Laboratory has actually measured 311 Btu at noon of a clear day in Pittsburgh.

The intensity of impingement on the surface, *X*, not normal to the sun's rays, decreases with the angle of incidence, *I*, which is a function of time; a part of the energy, *B*, is not absorbed, but is reflected away. The magnitude of *B* is a function of the character of the surface, *X*, and of the angle of reflection, *R*, which varies with the time of day. The remaining energy, C, is absorbed as heat by the surface, *X*, raises its temperature enough to cause the radiation, *D*, back through the air, and the convection loss, *E*, by direct contact with the air. Also, as the temperature of the surface, *X*, is raised by the absorption of radiant energy, heat flows in the direction, *F*. As the temperature of the surface, *X*, rises to a maximum at noon and then recedes, a wave of heat advances towards *F*. Because heat is required to raise the temperature of each increment of the distance, *X* to *Y*, the rate of heat flow past any point between *X* and *Y* diminishes as the wave penetrates through the structure, and when the wave reaches the surface, *Y*, it has a much lowered amplitude, dependent upon the conductivity, density, specific heat and thickness of the material in the wall, the film resistance of its lower surface, and the temperature of the air below. The crest of the wave reaching the surface, *Y*, will be delayed a certain time after the crest at the surface, *X*.

Conductivity, density and specific heat are factors which combine to damp out the wave amplitude. In a theoretical consideration of heat transfer they are combined into a single constant (h), called the diffusivity:

$$h = \sqrt{\frac{k}{c\rho}} \tag{1}$$

where

h = diffusivity
k = conductivity
c = specific heat
ρ = density

This combined constant takes into account the resistance to heat flow and the heat capacity of the structure.

The heat reaching the lower surface tends to build up its temperature, and a small amount of heat, *G*, returns in a reflected wave upward towards *X*. The magnitude of this reflected wave also depends upon the physical properties of the structure. Only *H*, the heat entering the air-conditioned space below, is of interest to the air-conditioning engineer.

An ideal solution of the problem of heat flow as affected by solar radiation would give the value of *H* for all hours of the day as a function of the intensity of solar radiation perpendicular to the direction of the sun's rays, of the atmospheric conditions, and of the physical characteristics of the structure. Such a solution involves the following factors: the intensity of solar radiation, *A*, perpendicular to the direction of the rays; the angle of incidence, *I*, of im-

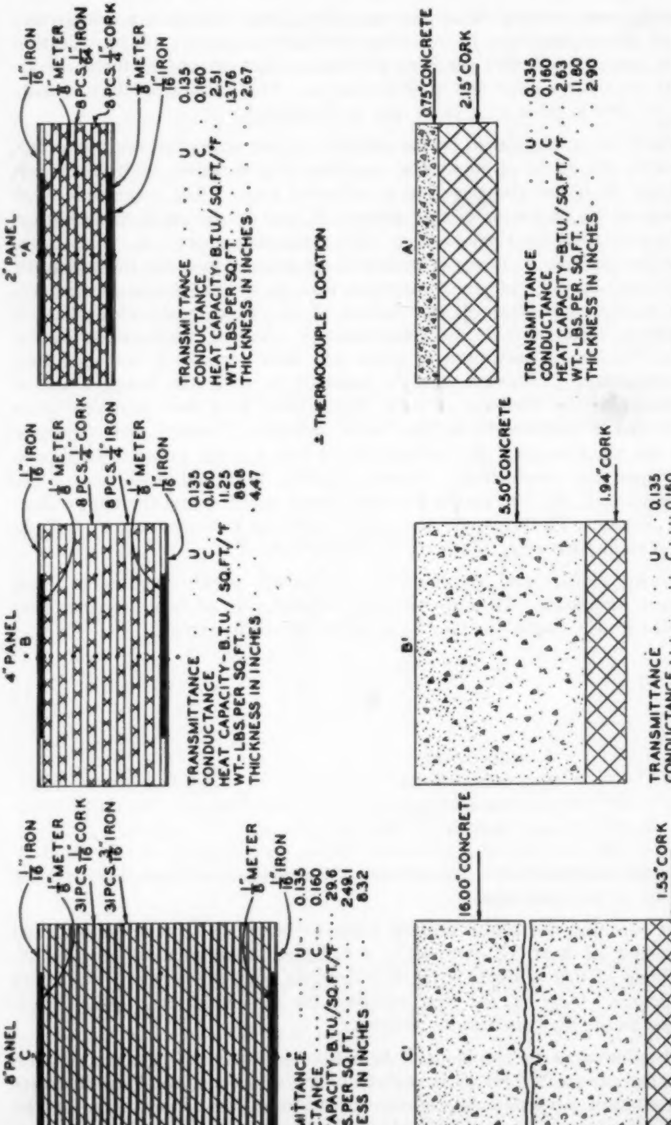


FIG. 2. IRON AND CORK PANELS WITH COMPARATIVE PANELS OF CONCRETE AND CORK

pingement against the surface; the reflected energy, B, depending upon the angle of reflection, R; the radiated heat, D, depending upon the convection from the surface, X; the difference in temperature of the surface, X, and the outside air; the temperature difference between the surfaces, X and Y; the conductivity, the density, specific heat and thickness of structure; the film resistance for the surfaces, X and Y; and the controlled temperature of the air below. This makes 13 factors upon which H depends, the first 5 of which are harmonic functions of time.

The problem of determining the value of H may be conveniently divided into two distinct parts, each of which is complicated: *first*, a determination of the varying temperatures of the surface, X, depending upon the values of A, B, C, D, E, and G, and the physical properties of the structure; and, *second*, a determination of the value of H at all times during the daily cycle, which value depends upon the change of the temperature of the surface, X, the physical properties of the structure, the film resistance of the inner surface, and the constant air temperature in the space below. Only the second of these two parts of the entire problem is treated in this paper, namely, the evaluation of H from the cyclic change in the temperature of the surface, X, and the characteristics of the structure XY.

ROOF PANELS STUDIED

At the beginning of the investigation it was considered desirable to study characteristics of heat penetration for three panels having approximately the same conductances but widely different heat capacities. Since it was desirable to eliminate as many variables as possible, and at the same time to have panels which could be considered homogeneous to the extent that the material in any small cube in the panel would be exactly similar to that of any other similar cube in the same panel, roof panels were built up by laminating alternate sheets of iron and cork, which afforded an opportunity to distribute thermocouples through the layers. Cross-sections of these panels, studied in the summer of 1930, are shown in Fig. 2. For convenience in comparing the iron and cork panels with structures familiar to the heating and ventilating engineer, the figure also shows panels of concrete and cork having approximately the same conductance and heat capacity.

During the summer of 1931, a 6-in. concrete, a 4-in. gypsum, and a 2-in. yellow pine panel were studied, as was also a panel made up of 3-in. concrete and 1-in. cork, tested first with concrete upwards, and later with the cork on top. (In the following references to this panel, the first term of the compound word, cork-concrete or concrete-cork, indicates the side of the panel which was upward during test.) All panels were 3 ft square. The concrete used was 4:2:1 mix of gravel, sand and cement with a 4½-in. slump. The concrete was poured onto smooth 1-in. corkboard at the bottom of the form so that concrete and cork were tightly bonded together and remained so throughout the tests. The gypsum panel was made of commercial roofing gypsum mixed according to specifications for a gypsum roof supplied by a large manufacturer. The concrete, the cork-concrete, and the gypsum panels were poured by Charles W. Larkin, Head Instructor of Masonry Construction at Carnegie Institute of Technology. These panels were perfect as regards uniformity of thickness and smoothness of surfaces. The yellow pine panel

was made by gluing together 2¼-in. x 6-in. strips of yellow pine and dressing them down to a panel accurately 2 in. in thickness with smooth planed surfaces.

The over-all physical properties of the 7 panels are given in Table 1. The over-all conductivity values were obtained either directly by testing the panels as placed in their set-up, or by averaging the known conductivities of their homogeneous component parts. The over-all densities and specific heats are

TABLE 1. PHYSICAL PROPERTIES OF PANELS

Description of Panel	Thickness l		Conductivity k		Density ρ		Specific Heat c	Inner Film Conductance, h_i	
	Eng.	Metric	Eng.	Metric $\times 10^{-4}$	Eng.	Metric		Eng.	Metric $\times 10^{-4}$
2" Laminated Iron & Cork	2.67	6.78	.427	1.470	61.68	.988	.192	1.0	1.355
4" Laminated Iron & Cork	4.47	11.35	.734	2.528	238.77	3.826	.128	1.25	1.694
8" Laminated Iron & Cork	8.32	21.13	1.231	4.240	357.05	5.722	.119	1.25	1.694
2" Pine	2.156	5.476	.830	2.858	37.46	.600	.467	1.9	2.574
6" Concrete	6.188	15.717	8.00	27.520	148.6	2.381	.230	1.9	2.574
4" Gypsum	4.188	10.637	1.445	4.971	64.89	1.040	.234	1.9	2.574
4" Cork-Concrete 4" Concrete-Cork	4.188	10.637	1.218	4.190	101.03	1.619	.218	1.9	2.874
All values based on average between outer and inner surfaces.									

averages for the entire individual panel as constructed. The densities and specific heats of the component homogeneous parts were determined by test, except in a few cases where they are handbook data. The physical properties of the homogeneous materials going into the various panels are given in Table 2.

TABLE 2. PHYSICAL PROPERTIES OF MATERIALS USED IN PANELS

Material	Conductivity k	Density ρ	Specific Heat, c
Iron Plate	33.3	489.5	.116
Gasket Cork	.35	14.43	.43
Pine	.89	37.45	.47
Concrete	15.0	148.6	.230
Gypsum	1.58	64.9	.234
Paper & Heat Meter	5.51 (a)	45.1	.35
Corkboard	.35	15.90	.43

TEST SET-UP

The test equipment for studying the effect of solar radiation on heat transfer was built on the flat roof of a low building at the Pittsburgh Station of the United States Bureau of Mines, so situated that at no time during the day was the roof without an unrestricted view of the sun. This set-up provided for the simultaneous study of three panels, each 3 ft. square, which formed a roof over an air space 3 ft. high by 3 ft. wide by 9 ft. long as shown in Fig. 3 and Fig. 4.

The floor and side walls of this chamber were insulated with 4 in. of corkboard, *A* tarred on the outside to prevent air infiltration and to furnish a weatherproof covering. The 3 test panels, *B, C, D,* form the roof of the air-conditioned space, and are so placed that their lower surfaces are in the same plane. The cork insulation on the sides of the air-conditioned space extended to the upper surfaces of the 3 panels so as to insulate the edges of the panels, and the joints between the panels and the insulation, and the joints between the edges of different panels were sealed with tar.

Provision for maintaining constant temperatures and humidities in the space below the roof panels was furnished by an ice-cold water spray, *E,* the well-distributed low-intensity 500 watt electric heater, *F,* and the thermostat, *G.* Cold water was pumped through insulated piping to the cooling spray by a

FIG. 3. TEST CHAMBER WITH THREE ROOF PANELS UNDER TEST

rotary gear pump from an insulated ice bath in the room below. The spray water was caught in the receiver, *H,* and returned to the ice bath by gravity. By varying the volume of spray water, the amount of cooling needed to maintain the air space at any predetermined temperature was readily controlled. The spray maintained the relative humidity of the conditioned air a constant, sufficiently low to prevent condensation on any of the interior surfaces. When heating was necessary, the thermostat, *G,* by regulation relays, controlled the current passing through the heater, *F.*

The thermostat held temperatures sufficiently constant for periods preliminary to the actual tests; for the accurate conditions necessary during tests, the heat was controlled manually by either graduating the current through the heater, *F,* or by adjusting the volume of cold water passing through the spray, using an orifice meter and hand-controlled valves. To prevent stratification of the air, a small slow speed blower, *J,* was located on the floor at one end of the chamber. The air inlet of this blower was restricted by a damper to allow

only a small volume of air to be moved at low velocity. This agitated the air in the box enough to prevent localized heating or cooling, but not sufficiently to cause an appreciable increase in heat transfer from the under surface of the panels.

Nicholls heat flow meters,[6] *L* and *O*, were firmly fastened to the lower and the upper surfaces of the panels, and their edges were sealed to the panels with adhesive tape. These meters gave the instantaneous flows of heat for both the upper and lower surfaces. The upper meters were covered with black oilcloth, *K*, to protect them from the weather. To obtain uniform, natural surfaces, the upper oilcloth and the lower meter surface were painted a dead black with lampblack pigment thinned with weak shellac.

Thermocouples of No. 36 gage B & S copper-constantan wire were fixed in

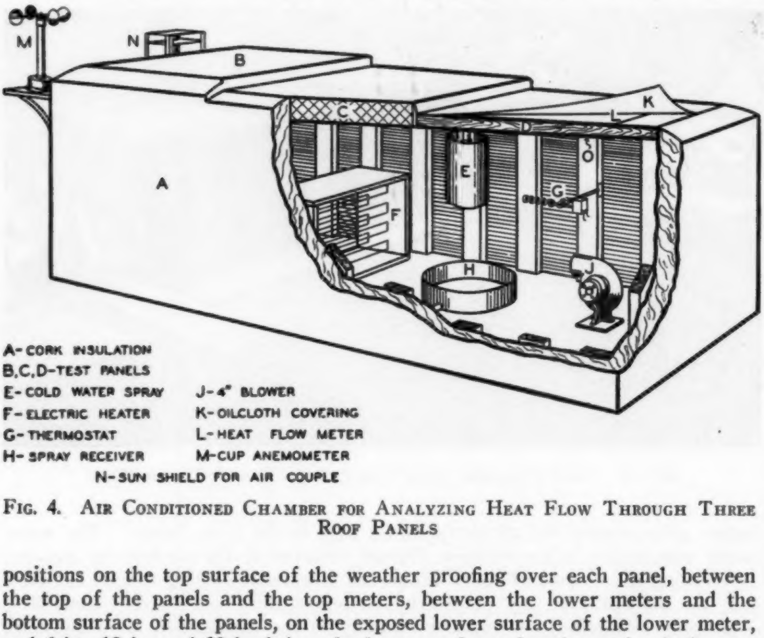


FIG. 4. AIR CONDITIONED CHAMBER FOR ANALYZING HEAT FLOW THROUGH THREE ROOF PANELS

positions on the top surface of the weather proofing over each panel, between the top of the panels and the top meters, between the lower meters and the bottom surface of the panels, on the exposed lower surface of the lower meter, and 6 in., 18 in. and 30 in. below the lower surface of each panel. A thermocouple for observing air temperatures in the sun was located 6 in. above the center of the middle panel; another thermocouple for observing shaded air temperatures was put under a double-deck shield, *N*, Fig. 4; and extra thermocouples were located at all important points to serve as checks. An electrical cup anemometer, *M*, placed near the top of the panels recorded all air movement over the face of the panels.

A pyrheliometer located near the set-up indicated the intensity of solar radiation throughout the day. This instrument, shown in the photograph,

[6] See Bibliography, *b*.

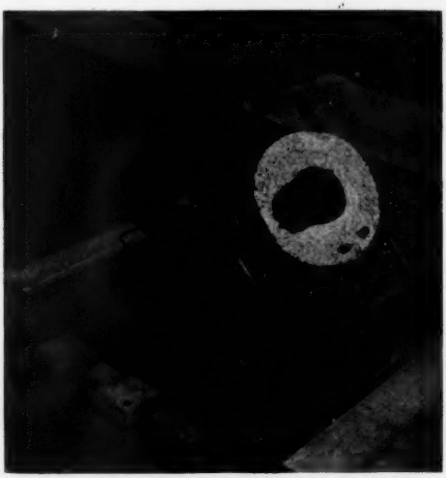


FIG. 5. PYRHELIOMETER FOR AUTOMATICALLY RECORDING INTENSITY OF SOLAR RADIATION

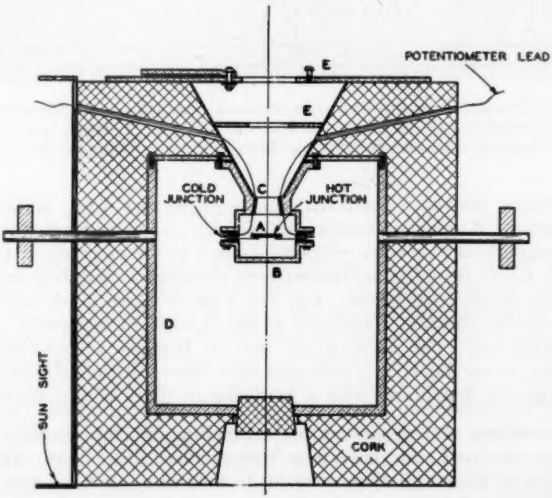


FIG. 6. CROSS SECTION OF THE LABORATORY PYRHELIOMETER

Fig. 5, and the drawing, Fig. 6, was designed and built at the A. S. H. V. E. Laboratory. It consisted of a smoked, dead-black, sensitive disk, *A*, containing 31 hot junctions of No. 40 B & S gage thermocouples, the cold junctions of which were located between the flanged joint of the cup, *B*. When the instrument was sighted on the sun, the sun shone onto the sensitive disk through the orifice, *C*, which was so designed that the sun entirely covered the thermocouple junctions in the disk, even when the instrument failed to point directly at the sun by an angle of 2 deg. During a day, the cold junctions were held at practically a constant temperature because the insulated brass container, *D*, was filled with water, which has great heat capacity. Small changes in the temperature of the cold junctions did not affect readings of the instrument, since the temperature difference between the hot and cold junctions remained constant.

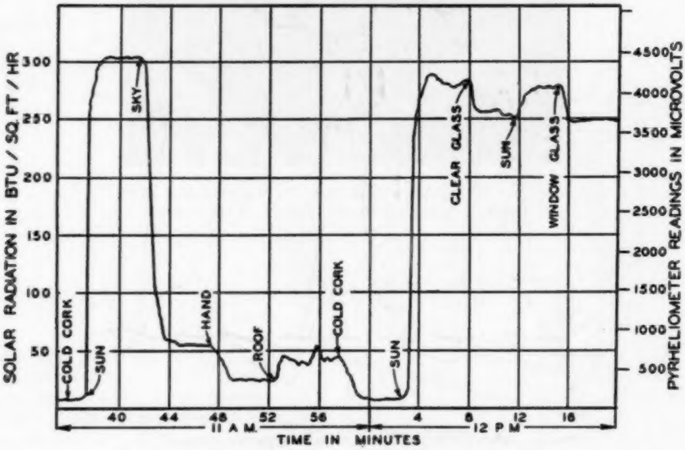


FIG. 7. CURVE SHOWING PYRHELIOMETER INDICATION AND RADIATION INTENSITY FROM THE SUN, SKY, A PERSON'S HAND, A ROOF, AN ICE COLD SURFACE AND THE RADIATION TRANSMISSION THROUGH GLASS

The diaphragm shields, *E*, protected the sensitive disk from air currents and stray radiation. A trunnion mounting timed by a clock controlled the direction of the instrument so that when properly sighted in the morning, it followed the sun all day. Leads from the instrument went to either a recording or a precision potentiometer in the room below. Fig. 7 is a curve which gives some radiations read with the instrument. The scale at the right is in microvolts as recorded by the instrument. The scale at the left, in Btu per square foot per hour perpendicular to the sun's rays, corresponds to the record in microvolts obtained by calibrating the instrument with a Smithsonian silver disk pyrheliometer.

The instruments and apparatus for controlling air temperature within the chamber on the roof, for observing temperature, heat flows, wind velocity and intensity of solar radiation, were all located in a room directly below, and are shown in Fig. 8.

This set-up allowed 3 panels to be tested simultaneously with heat flow meters and thermocouples so arranged that at any time readings could be had of the instantaneous heat flow into the top and out of the bottom surfaces, and of the temperature gradients through the panels subjected to varying weather conditions at the outer surface and constant air temperature below.

TEST OPERATION AND RESULTS

Time observations were made in mean solar time, which for the latitude and longitude of Pittsburgh and for the days studied is 20 to 26 min behind Eastern

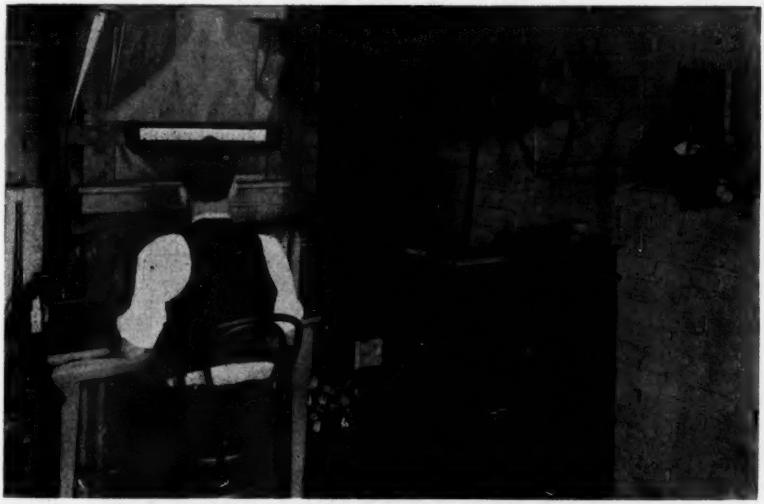


FIG. 8. APPARATUS AND INSTRUMENTS USED TO CONTROL TEMPERATURES AND TO OBSERVE TEMPERATURES, HEAT FLOWS, WIND VELOCITIES AND INTENSITY OF SOLAR RADIATION IN THE TEST SET-UP ON THE ROOF ABOVE

Standard Time. All calculations involving time, and all references made to time, are in terms of mean solar, or sun time.

All tests were run continuously for 24 hr starting shortly before sunrise. If no test had been run the day before, the air temperature in the chamber was controlled at approximately 70 F for a preliminary time of 12 hr. This was done automatically by thermostatic control. During the 24 hr of the test period, the air temperature in the chamber was controlled manually at 69.6 F ± 0.2 deg. Half-hourly observations were made of the heat flow as given by the 6 heat meters, of the temperatures indicated by the thermocouples, of the wind velocity, the intensity of solar radiation, and the general weather conditions. Whenever weather conditions permitted, tests were run continuously day after day. However, it was found that comparatively few days were sufficiently free from clouds or haze to permit the collection of satisfactory data.

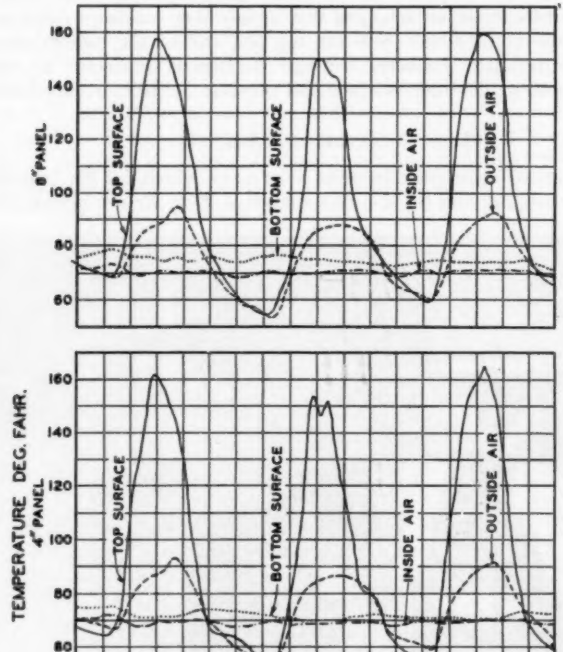


FIG. 9. CURVES SHOWING OBSERVED TEMPERATURES OF TOP AND BOTTOM SURFACES AND INSIDE AND OUTSIDE AIR FOR 2 IN., 4 IN., AND 8 IN. IRON AND CORK PANELS

The data used for analysis in this paper include the top surface temperature of the panel exposed to the air, the inner air temperature—always kept at 69.6 F—and the heat flow observed by the lower heat meter. To avoid errors introduced by edge losses, all temperatures were read at the center of the 18-in. square test section covered by the recording element of the heat flow meter.

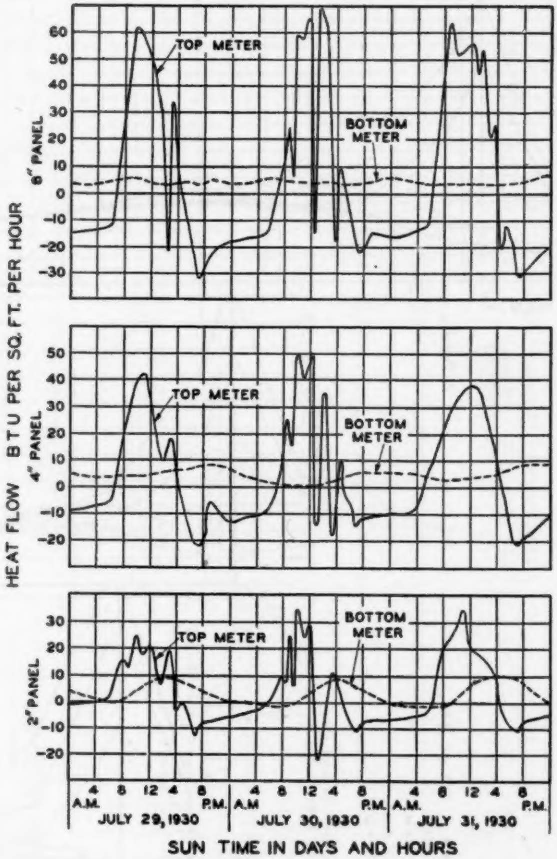


Fig. 10. Curves Giving Observed Heat Flows Through Top and Bottom Surfaces for 2 in., 4 in. and 8 in. Iron and Cork Panels

Fig. 9 shows the observed temperatures of the top and bottom surfaces of the 2-in., 4-in., and 8-in. laminated iron-cork panels and the inside and outside air temperatures; the 4 temperatures are plotted against time during 3 days, July 29, 30, and 31. Fig. 10 is a graph of the observed heat flows at the upper and lower surfaces for the same panels on the same days. Temperature gradients for one day through the iron and cork panels, as measured, are plotted in Fig. 11 for various times between midnight and noon, and are continued in

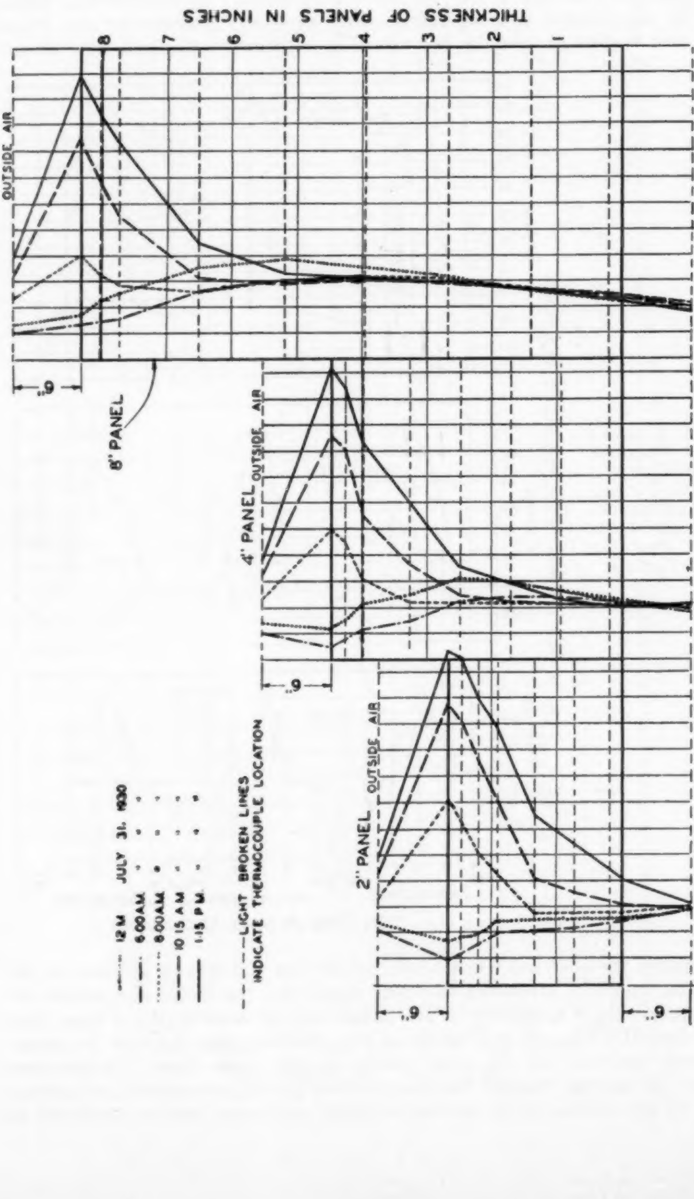


FIG. 11. TEMPERATURE GRADIENTS AT VARIOUS TIMES FROM MIDNIGHT TO NOON FOR 2-IN., 4-IN., AND 8-IN. IRON AND CORK PANELS

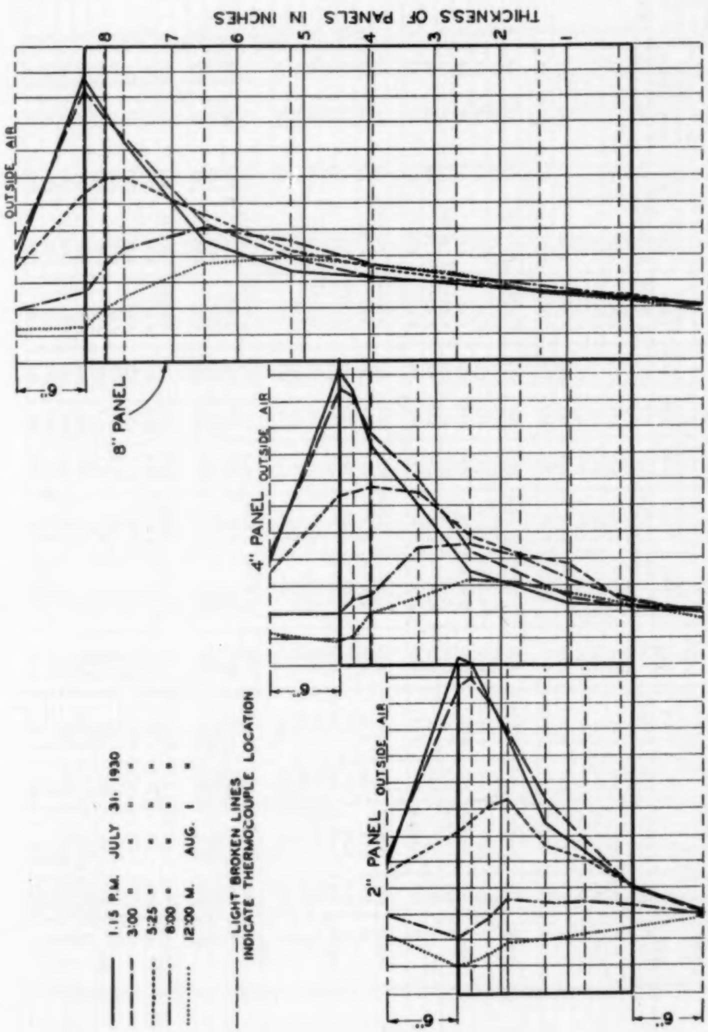


FIG. 12. TEMPERATURE GRADIENTS AT VARIOUS TIMES FROM NOON TO MIDNIGHT FOR 2-IN., 4-IN., AND 8-IN. IRON AND CORK PANELS

TABLE 3. CHARACTERISTICS OF TOP SURFACE TEMPERATURES AND LOWER SURFACE HEAT FLOWS

PANEL	DATE	OUTSIDE AIR TEMPERATURE				OUTER SURFACE TEMPERATURE						INNER SURFACE HEAT FLOW					LAG BETWEEN CREST OF TEMP. AND CREST OF HEAT FLOW IN HOURS
		MAX.		MIN.													
	1930	°F.	TIME	°F.	TIME	BEGAN TO RISE A.M.	CREST TIME	TROUGH TIME	CREST TEMP.	TROUGH TEMP.	TEMP. RANGE	CREST TIME	TROUGH TIME	CREST BTU	TROUGH BTU	BTU RANGE	
2" Iron - Cork	July 29	89.8	3:48 P.M.	68.0	5:45 A.M.	5:40	12:15 P.M.	5:10 A.M.	160.5	57.5	103.0	2:30 P.M.	6:30 A.M.	9.0	0.0	9.0	2 Hrs. 15 Min.
" " "	" 30	87.0	2:15 P.M.	53.6	5:10 "	5:00	12:15 "	4:50 "	156.0	43.5	112.5	4:00 "	7:30 "	9.0	2.5	6.5	3 " 0 "
" " "	" 31	85.5	2:27 "	59.4	5:13 "	5:10	12:50 "	5:10 "	162.0	49.5	112.5	4:30 "	6:30 "	10.5	3.0	7.5	3 " 40 "
" " "	Sept. 9	72.0	11:35 A.M.	50.0	5:06 "	5:00	12:00 "	5:00 "	159.3	42.5	116.8	3:09 "	5:10 "	11.0	-3.5	14.5	3 " 09 "
" " "	" 10	94.0	1:10 P.M.	51.0	5:20 "	5:25	12:30 "	5:25 "	153.0	40.2	112.8	2:30 "	7:00 "	11.2	-3.5	14.7	2 " 0 "
" " "	" 18	92.0	1:22 "	59.0	5:35 "	6:05	1:25 "	5:25 "	133.5	38.8	94.7	2:20 "	3:40 "	10.3	-4.5	14.8	0 " 55 "
2" Iron - Cork	" 19	88.0	11:34 A.M.	54.0	7:09 "	7:15	1:00 "	5:50 "	135.3	44.2	91.1	3:00 "	1:00 "	6.9	-3.4	10.3	2 " 0 "
4" Iron - Cork	July 29	89.8	3:48 P.M.	68.0	5:45 A.M.	4:30	11:30 A.M.	4:30 A.M.	162.5	64.5	98.0	9:00 P.M.	4:30 A.M.	8.0	3.5	4.5	9 " 30 "
" " "	" 30	87.0	2:15 "	53.6	5:10 "	5:30	11:30 "	5:00 "	153.0	42.0	111.0	8:00 "	11:30 "	5.5	0.5	5.0	8 " 30 "
" " "	" 31	85.5	2:27 "	59.4	5:13 "	5:00	1:15 P.M.	5:00 "	164.5	44.0	120.5	10:00 "	9:00 "	8.5	3.0	5.5	8 " 45 "
" " "	Sept. 9	72.0	11:35 A.M.	50.0	5:05 "	5:00	12:40 "	5:00 "	149.1	46.0	103.1	5:40 "	8:40 "	6.5	1.7	4.8	5 " 0 "
" " "	" 10	94.0	1:10 P.M.	51.0	5:20 "	5:50	12:50 "	5:30 "	147.7	44.1	103.6	6:40 "	12:50 P.M.	5.2	-1.0	6.2	5 " 40 "
" " "	" 18	92.0	1:22 "	59.0	5:35 "	6:00	12:40 "	6:00 "	135.0	44.8	90.2	8:20 "	5:30 "	4.8	3.3	1.5	
4" Iron - Cork	" 19	88.0	11:34 A.M.	54.0	7:09 "	6:35	12:00 "	5:50 "	129.0	47.4	81.6	6:40 "	2:50 "	4.7	1.1	5.8	6 " 40 "
8" Iron - Cork	July 29	89.8	3:48 P.M.	68.0	5:45 A.M.	4:50	11:20 A.M.	4:50 A.M.	157.0	68.5	88.5	9:30 A.M.	7:00 P.M.	5.5	3.5	2.0	
" " "	" 30	87.0	2:15 "	53.6	5:10 "	4:15	12:00 P.M.	4:15 "	149.5	44.5	105.0	6:30 "	6:00 "	6.0	3.5	2.5	19 " 10 "
" " "	" 31	85.5	2:27 "	59.4	5:13 "	5:30	1:00 "	5:00 "	159.5	49.0	110.5	12:00 P.M.	4:00 "	7.0	3.5	3.5	24 " 0 "
" " "	Sept. 9	72.0	11:35 A.M.	50.0	5:05 "	5:30	1:05 "	5:30 "	146.5	48.6	97.9	12:22 "	5:00 "	4.8	3.3	1.5	
" " "	" 10	94.0	1:10 P.M.	51.0	5:20 "	5:35	12:19 "	5:45 "	144.0	47.6	96.4	8:40 A.M.	1:34 "	4.5	2.0	2.5	15 " 35 "
" " "	" 18	92.0	1:22 "	59.0	5:35 "	5:57	1:00 "	5:57 "	132.8	50.2	82.6	10:30 "	8:20 "	7.7	3.2	4.5	
8" Iron - Cork	" 19	88.0	1:22 "	54.0	5:35 "	7:18	1:25 "	7:00 "	127.0	50.1	66.9	6:20 "	4:51 "	6.5	-0.9	7.4	17 " 20 "
	1931																
2" Pine	July 25	86.6	2:30 P.M.	56.2	3:25 A.M.	5:00	12:50 P.M.	3:40 A.M.	146.0	51.5	94.5	2:00 P.M.	6:00 A.M.	25.0	-5.0	30.0	1 " 10 "
" "	" 26	94.4	1:20 "	56.1	4:00 "	4:50	12:40 "	3:40 "	153.5	50.7	102.8	1:40 "	5:20 "	26.0	-4.8	30.8	1 " 0 "
" "	" 27	97.5	1:45 "	60.8	4:30 "	5:10	12:30 "	3:00 "	152.5	56.0	96.5	2:10 "	5:20 "	28.5	-5.2	30.7	1 " 40 "
2" Pine	" 28	96.6	3:15 "	65.5	3:45 "	4:30	12:15 "	4:10 "	147.0	60.2	86.8	2:15 "	2:00 "	24.0	-3.8	27.8	2 " 0 "
6" Concrete	July 25	86.6	2:30 P.M.	56.2	3:25 A.M.	5:00	1:30 P.M.	4:30 A.M.	125.0	60.0	65.0	5:00 P.M.	7:00 A.M.	38.2	-5.0	43.2	3 " 30 "
" "	" 26	94.4	1:20 "	56.1	4:00 "	5:00	12:45 "	4:45 "	125.8	60.3	65.5	4:36 "	6:50 "	37.5	-4.0	41.5	3 " 50 "
" "	" 27	97.5	1:45 "	60.8	4:30 "	5:00	1:50 "	5:00 "	128.5	65.0	63.5	4:50 "	6:30 "	40.8	0.2	40.6	3 " 0 "
" "	" 28	96.6	3:15 "	65.5	3:45 "	5:30	1:40 "	5:00 "	125.3	68.3	57.0	4:15 "	6:30 "	41.0	1.2	39.8	2 " 35 "
" "	Sept. 7	82.2	1:30 "	50.5	4:35 "	5:50	1:20 "	5:25 "	118.2	55.2	63.0	4:05 "	7:15 "	30.5	-6.5	37.0	2 " 45 "
" "	" 8	82.8	3:30 "	51.7	6:00 "	6:00	3:00 "	5:15 "	96.0	55.0	41.0	5:00 "	7:20 "	15.0	-9.0	24.0	2 " 0 "
" "	" 9	92.9	1:30 "	56.4	1:30 "	5:30	1:15 "	5:00 "	116.0	59.5	56.5	4:30 "	5:40 "	28.0	-6.0	34.0	3 " 15 "
" "	" 10	97.9	3:00 "	65.5	5:15 "	5:40	1:30 "	4:50 "	117.0	66.5	60.5	4:10 "	7:20 "	29.0	-5.0	29.5	2 " 40 "
6" Concrete	" 11	95.9	3:00 "	64.7	5:15 "	5:45	1:10 "	5:00 "	120.0	65.0	55.0	4:00 "	7:10 "	29.5	-2.0	31.5	2 " 50 "

4" Gypsum	Sept. 7	82.2	1:30 P.M.	50.5	4:35 A.M.	5:30	1:00 P.M.	5:30 A.M.	138.5	46.0	92.5	3:15 P.M.	7:00 A.M.	18.0	-5.8	23.8	2 · 15 ·
" "	" 8	82.8	3:30 " "	51.7	6:00 " "	5:45	3:20 " "	5:30 " "	116.0	48.0	68.0	5:00 " "	6:45 " "	9.5	-6.2	15.7	1 · 40 ·
" "	" 9	92.9	1:30 " "	56.4	1:30 " "	6:00	1:00 " "	1:50 " "	132.4	53.5	78.9	4:10 " "	5:00 " "	16.5	-3.2	19.7	3 · 10 ·
" "	" 10	97.9	3:00 " "	65.5	5:15 " "	5:50	1:35 " "	5:30 " "	132.0	61.5	70.6	4:00 " "	7:00 " "	18.5	-1.4	19.9	2 · 25 ·
4" Gypsum	" 11	95.9	3:00 " "	64.7	5:15 " "	6:30	1:00 " "	5:00 " "	140.6	60.6	80.1	4:20 " "	7:00 " "	8.3	-3.0	11.3	3 · 20 ·
4" Cork-Concrete	Sept. 7	82.2	1:30 P.M.	50.5	4:35 A.M.	6:00	12:40 P.M.	5:30 A.M.	139.0	43.0	96.0	4:00 P.M.	7:30 A.M.	12.0	-1.8	13.8	3 · 20 ·
" " "	" 8	82.8	3:30 " "	51.7	6:00 " "	5:40	2:50 " "	5:20 " "	121.0	45.0	76.0	5:10 " "	6:30 " "	6.4	-3.9	10.3	2 · 20 ·
" " "	" 9	92.9	1:30 " "	56.4	1:30 " "	2:00	12:35 " "	2:00 " "	135.8	51.8	84.0	4:10 " "	5:30 " "	11.8	-2.8	14.8	3 · 35 ·
" " "	" 10	97.9	3:00 " "	68.5	5:15 " "	5:00	1:00 " "	4:30 " "	136.5	60.0	76.5	4:00 " "	7:00 " "	12.0	-1.0	13.0	3 · 0 ·
4" Cork-Concrete	" 11	95.9	3:00 " "	64.7	5:15 " "	6:00	12:50 " "	4:30 " "	141.0	59.2	81.8	4:10 " "	6:00 " "	12.0	-1.0	13.0	3 · 20 ·
4" Concrete-Cork	July 25	86.8	2:30 P.M.	56.2	3:25 A.M.	5:00	2:00 P.M.	4:50 A.M.	137.5	55.2	82.3	4:05 P.M.	6:30 A.M.	15.9	-2.7	18.6	2 · 05 ·
" " "	" 26	94.4	1:20 " "	56.1	4:00 " "	5:30	1:50 " "	5:00 " "	142.0	57.0	85.0	4:00 " "	5:40 " "	15.8	-3.0	18.8	2 · 10 ·
" " "	" 27	97.5	1:45 " "	60.8	4:30 " "	5:20	2:00 " "	5:20 " "	141.0	63.2	77.8	4:00 " "	6:00 " "	17.5	-1.2	18.7	2 · 0 ·
4" Concrete-Cork	" 28	96.6	3:15 " "	65.5	3:45 " "	5:30	1:50 " "	5:10 " "	137.0	67.0	70.0	3:00 " "	6:40 " "	16.9	-0.2	17.1	1 · 10 ·

TABLE 4. OBSERVED AND CALCULATED HEAT FLOW

Time	No. of Hours	HEAT FLOW															
		2" Pine July 27, 1931		6" Concrete Sept. 10, 1931		Gypsum Sept. 10, 1931		Cork-Concrete Sept 10, 1931		Concrete-Cork July 27, 1931		2" Iron-Cork July 31, 1930		4" Iron-Cork July 31, 1930		8" Iron-Cork July 31, 1930	
		Obs.	Calc.	Obs.	Calc.	Obs.	Calc.	Obs.	Calc.	Obs.	Calc.	Obs.	Calc.	Obs.	Calc.	Obs.	Calc.
6-7 A.M.	1	-5.5	0.6	.0	-1.4	-1.0	-0.6	0	-0.5	-1.5	0.5	-1.7	-0.7	2.4	-0.8	3.0	-0.7
9-10 " "	1	12.0	4.4	2.5	11.1	1.0	5.1	1.0	4.4	2.5	3.6	2.5	1.6	1.5	1.6	3.4	1.5
12-1 P.M.	1	23.0	6.4	18.0	13.9	11.0	6.3	8.5	5.6	11.5	5.2	8.5	2.5	2.0	2.6	3.5	2.4
3-4 " "	1	24.5	6.5	28.5	15.0	17.0	6.8	11.5	6.0	17.5	5.3	10.5	2.6	4.7	2.7	3.7	2.5
6-7 " "	1	12.5	3.3	25.0	8.4	13.5	3.8	10.0	3.4	14.0	2.7	8.0	0.7	6.5	0.8	3.8	0.7
9-10 " "	1	1.5	0.9	15.0	1.6	4.0	0.7	5.0	0.6	7.5	1.8	2.0	-0.1	7.0	-0.1	4.0	-0.1
12-1 A.M.	1	-3.0	-0.4	7.5	-1.2	1.0	-0.6	2.0	-0.5	2.0	-0.4	-1.0	-0.6	6.0	-0.7	4.5	-0.6
3-4 " "	1	-4.0	-1.1	1.5	-2.4	-1.0	-1.1	.0	-1.0	0.5	-0.9	-1.4	-1.2	4.0	-1.2	4.0	-1.1
9 A.M.-5 A.M.	8	153.5	52.0	146.3	104.0	81.6	47.2	60.2	41.6	88.7	42.4	65.1	19.0	23.3	19.4	24.1	17.8
8 P.M.-11 P.M.	3	5.6	2.9	48.0	4.6	19.2	2.1	18.7	1.8	15.7	1.6	4.1	0.4	14.1	-2.7	10.5	-0.3

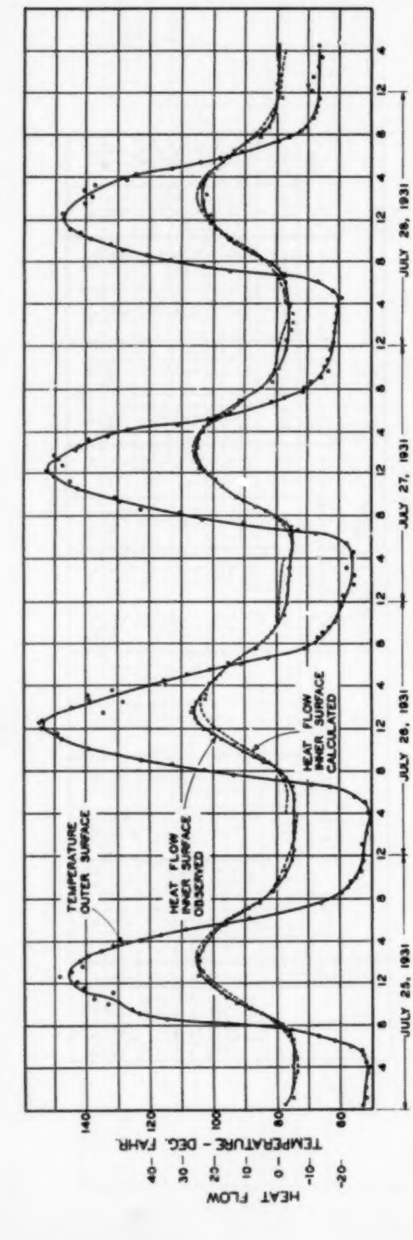


FIG. 13. CURVES FOR 2-IN. PINE PANEL

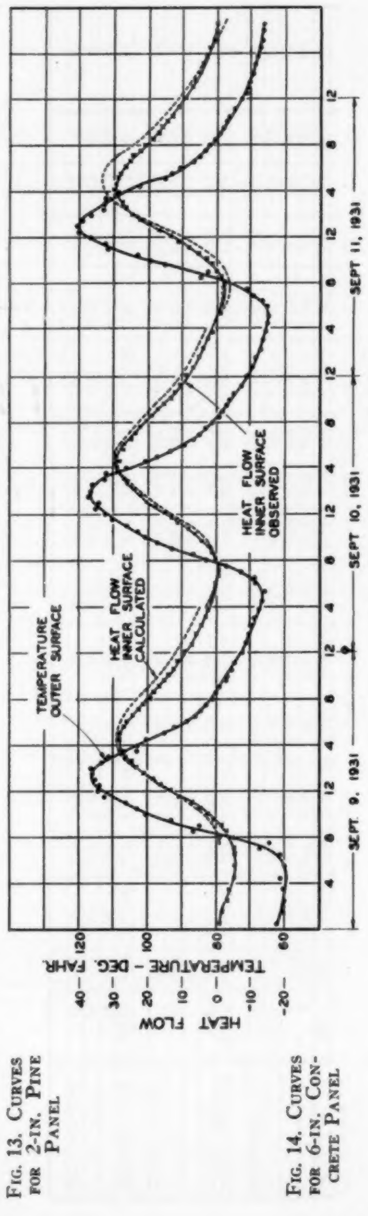


FIG. 14. CURVES FOR 6-IN. CONCRETE PANEL

Fig. 12 between noon and the following midnight. These give an interesting series of temperature gradient curves from the air above through the panel to the air below, which show just how the temperature wave progresses and how its amplitude is damped out as it penetrates to greater depths.

Fig. 13 gives the top surface temperature and the inner surface heat flow as observed on the pine panel for 4 days, July 25 to 28 inclusive. The result of a mathematical analysis of the heat flow through the lower surface is also given which will be discussed later. Similar data are given for 3 days on the concrete panel in Fig. 14. Figs. 15 and 16 give the solar radiation normal to the sun and normal to a horizontal surface for the days shown in Figs. 13 and 14. The intensity of solar radiation in Btu per square foot per hour on a surface perpendicular to the direction of the rays of the sun were obtained from the recorded chart of the A. S. H. V. E. Laboratory pyrheliometer, and the intensity of solar radiation on a horizontal surface was calculated from the recorded curve. Data similar to that in Figs. 13 and 14 are given for 5 days on the gypsum panel in Fig. 17, for 3 days on the cork-concrete (cork up) panel in Fig. 18, for 2 non-consecutive days on the concrete-cork (concrete up) panel in Fig. 19, and for one day each on the 2-in. and 4-in. iron-cork panels in Figs. 20 and 21. Figs. 22 and 23 give the temperatures of the top surface exposed to the atmosphere, the temperatures below the upper meter, the air temperatures above the panels, the temperatures of the lower surface exposed to the conditioned air, the heat flows through the lower surface, and other calculated curves which will be discussed later, for one day each on pine and concrete respectively. Figs. 24, 25 and 26 give the top surface temperature and the lower surface heat flows for an additional day each for pine, concrete and gypsum respectively.

The curves in Figs. 9 to 26 show that immediately after sunrise the top surface temperature begins to rise and that it continues rising until shortly after noon, when the temperature begins to recede rapidly until after sunset, from which point it continues to fall slowly until sunrise the next morning. The part of the curve from sunrise to sunset is a close approximation of a half cycle of a simple harmonic curve. The part of the curve from sunset to sunrise does not represent the other half cycle of the simple harmonic curve, but is shorter in time, because during the time of year represented by the data, nights are shorter than days. The amplitude of the night part of the cycle is not as great as for the day, and does not reach its trough midway between sunset and sunrise, but at a time very shortly before sunrise. This failure of the night half of the curve to reach a trough with amplitude equal to that at noontime may be accounted for by the lack of negative solar radiation at night. The heat flow through the lower surface gives a curve similar in shape to the top surface temperature, but lagging behind it by an amount depending upon the panel studied. Table 3 gives timing and characteristics of the top surface temperatures and the lower surface heat flows for different days and panels.

Comparisons between the calculated and observed heat flows for the 8 panels are shown for different periods of the day in Table 4. The calculated heat flows were based on the conductivities and inner film conductances given in Table 1, an outer film conductance of 2.0, and the average difference in temperature between the inside and outside air. Direct comparison between panels can be made only for those studied on the same day. It will be observed that for

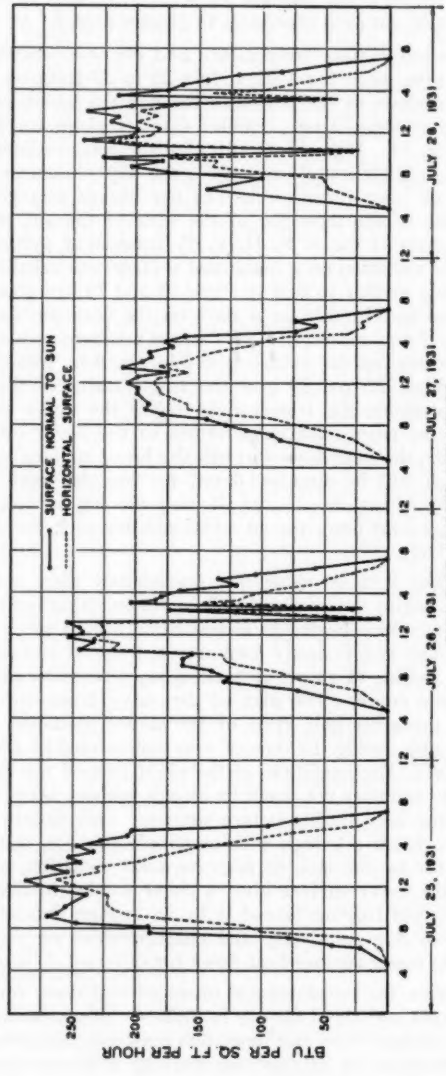


FIG. 15. CURVES GIVING SOLAR INTENSITY NORMAL TO SUN AND ON HORIZONTAL SURFACE

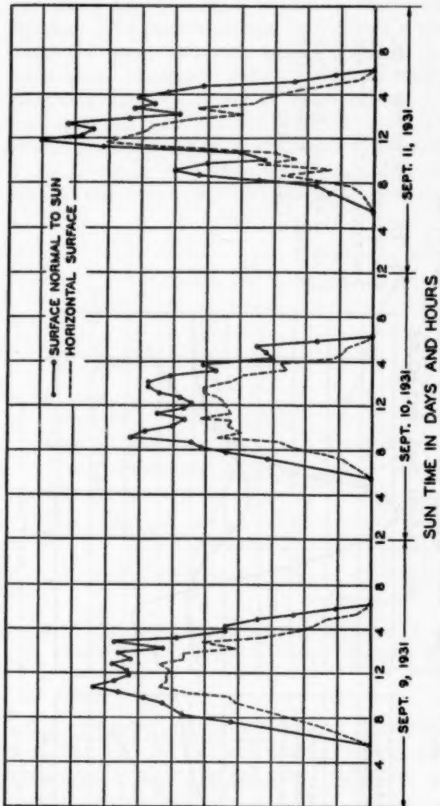


FIG. 16. (RIGHT) CURVES GIVING SOLAR INTENSITY NORMAL TO SUN AND ON HORIZONTAL SURFACE

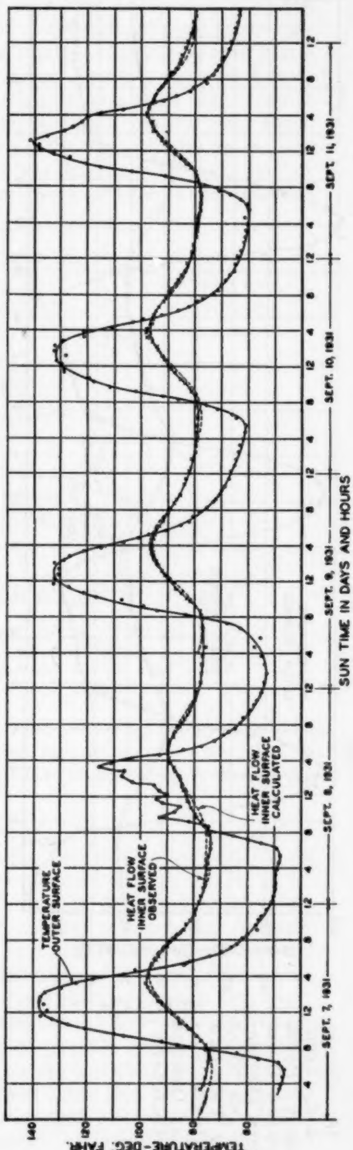


FIG. 17. (BELOW) CURVES FOR 4-IN. GYPSUM PANEL SHOWING OUTER SURFACE TEMPERATURE AND HEAT FLOW THROUGH THE INNER SURFACE AS OBSERVED AND CALCULATED

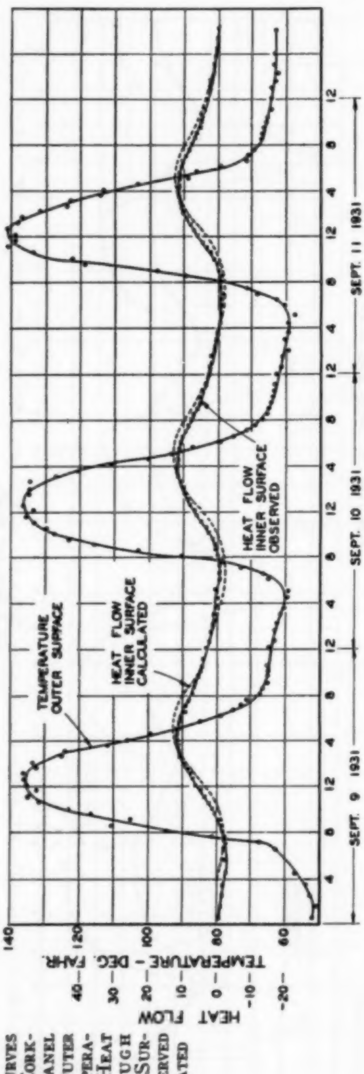


FIG. 18. CURVES FOR 4-IN. CORK-CONCRETE PANEL SHOWING OUTER SURFACE TEMPERATURE AND HEAT FLOW THROUGH THE INNER SURFACE AS OBSERVED AND CALCULATED

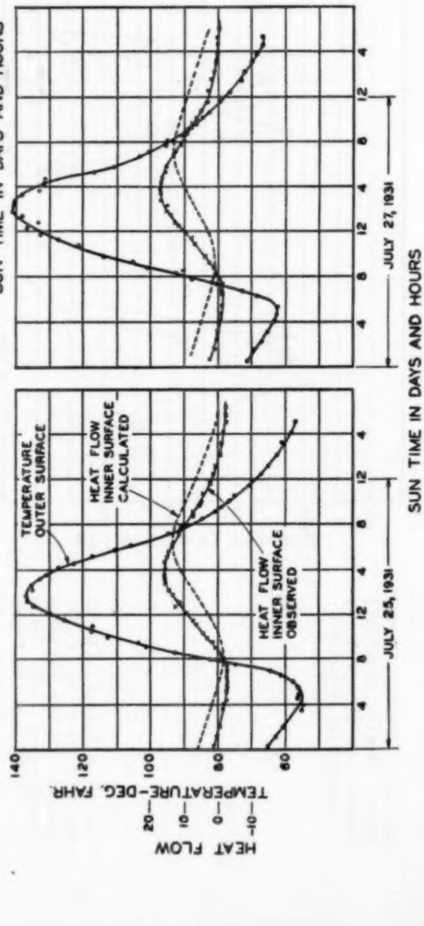


FIG. 19. CURVES FOR 4-IN. CONCRETE-CORK PANEL SHOWING OUTER SURFACE TEMPERATURE AND HEAT FLOW THROUGH THE INNER SURFACE AS OBSERVED AND CALCULATED

any panel the heat flow reaches a maximum sometime after noon and then recedes. The calculated heat flow shows a similar cycle, though not necessarily timed the same, and with widely different magnitudes. All panels give maximum heat penetration into the interior during the period, 3 P.M. to 4 P.M., excepting those of iron-cork construction, which give maximum rates from 3 P.M. to 9 P.M. for the 2-in. and 4-in., and after midnight for the 8-in. panel. It will be observed that the calculated heat flow for the day and evening periods, 9 A.M. to 5 P.M., and 8 P.M. to 11 P.M., is always less than the observed heat flow for the same periods. At some times this discrepancy amounts to an observed

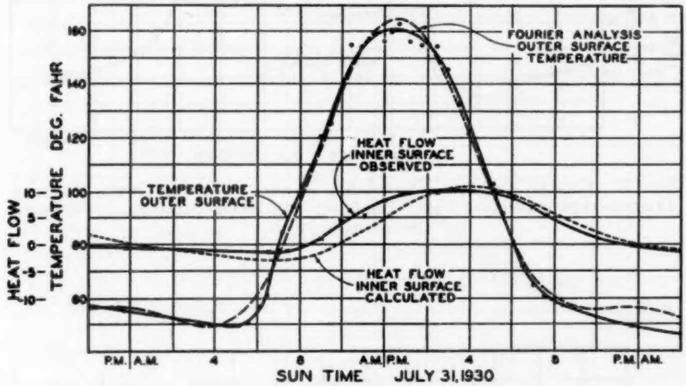


Fig. 20. Curves for 2-in. Iron-Cork Panel Showing Outer Surface Temperature and Heat Flow Through the Inner Surface as Observed and Calculated

value of approximately 3 times the calculated value for the 2-in. pine and the 2-in. iron and cork panels during the day period, and to an observed value 10 times greater than the calculated value for the 6-in. concrete panel during the evening period.

Since the value of the conductivity is important in the analysis of the flow of heat, its accurate determination is necessary. For the earlier part of the work, the conductivities used were taken from accepted data in the A. S. H. V. E. Guide, but it was soon apparent that a check on these evaluated values would be desirable. By holding the air in the chamber at a constant temperature and covering the upper surface with a deep bed of 32 F slush ice and water, an excellent determination of the conductivities of the panels was made with them in place. The calibrated Nicholls heat flow meters on the upper and lower surfaces gave the heat flow into and out of the lower and upper surfaces of the panels, in addition to a check of possible heat losses through the edges. A frame, 6 in. deep, was built up on the outer edges of the panels to hold the ice in position. The respective heat flows and the temperatures of the different surfaces of the panels were recorded when the rate of flow became constant. The tests for the determination of conductances and conductivities were continued for a length of time sufficient to assure a balanced

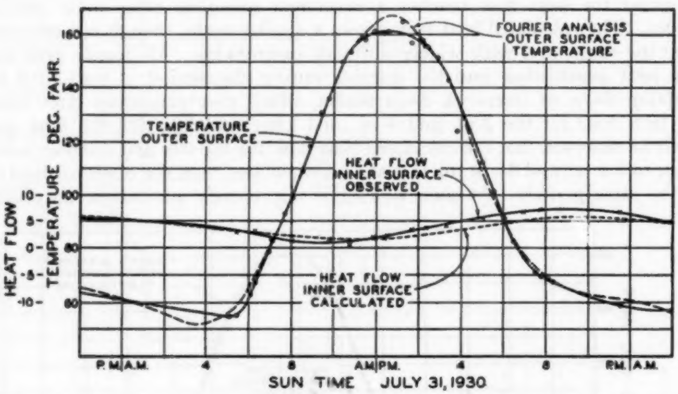


FIG. 21. CURVES FOR 4-IN. IRON-CORK PANEL SHOWING OUTER SURFACE TEMPERATURE AND HEAT FLOW THROUGH THE INNER SURFACE AS OBSERVED AND CALCULATED

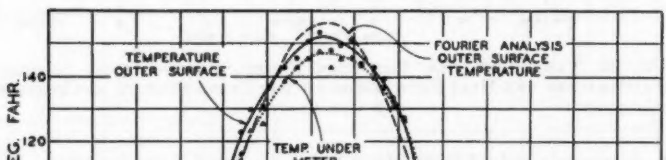


FIG. 22. CURVES FOR 2-IN. PINE PANEL SHOWING TEMPERATURES AND HEAT FLOWS AS OBSERVED AND CALCULATED

steady state condition of heat flow. From the temperatures, rates of heat flow, and thickness of the panel, it was a simple matter to work out the conductivities by the ordinary straight line flow formulae. These conductivities were of particular advantage, as they were run only a few days before the cyclic tests were made; consequently there was little time for the conductivities to change by aging or drying.

The specific heats used for the first wave flow calculations were originally handbook data, but these were later checked by a rather unique method developed at the Laboratory. A pulverized sample of the material held in a sealed glass

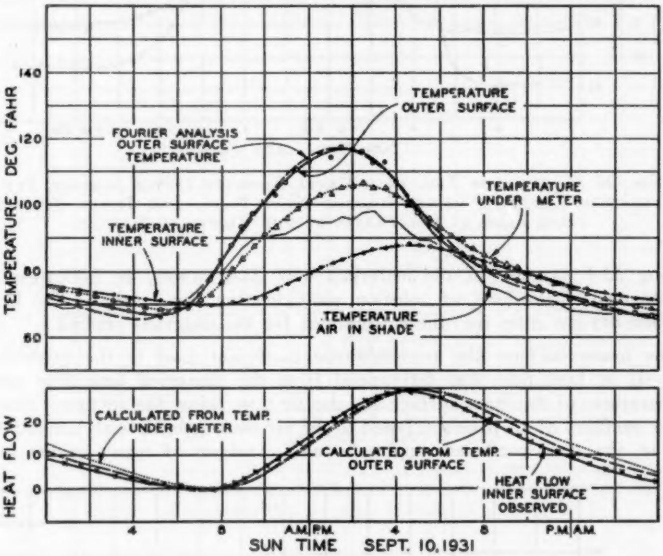


FIG. 23. CURVES FOR 6-IN. CONCRETE PANEL SHOWING TEMPERATURES AND HEAT FLOWS AS OBSERVED AND CALCULATED

test tube was cooled to a constant temperature of 32 F by completely covering it with slush ice over night. Water at a temperature slightly above that of the room was kept agitated in the cup of a student calorimeter while the cooled sample of material was quickly added to the cup. All masses and temperatures were chosen so that the resulting mixture had a temperature the same as that of the room. Temperature observations were made of the mixture of sample and water with an accurate thermometer, and curves for cooling corrections were drawn. The specific heat of the unknown sample was determined from the weight, temperatures and constants for the calorimeter by the usual calculations. The fact that the sample was cooled instead of heated before being mixed with the water in the calorimeter greatly added to the accuracy of the method, since the ice bath gave a very constant temperature, insuring an accurate temperature determination of the sample which would be otherwise hard to get. By

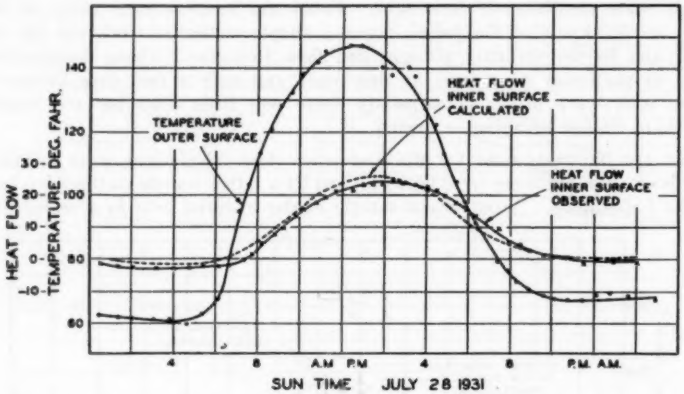


FIG. 24. CURVES FOR 2-IN. PINE PANEL SHOWING OUTER SURFACE TEMPERATURE, OBSERVED INNER SURFACE HEAT FLOW, AND INNER SURFACE HEAT FLOW AS CALCULATED BY THE EMPIRICAL SOLUTION

mixing 32-F samples of the materials with 32-F water, an indication could be obtained if the heat of solution was affecting the results. The heat of solution did not enter into the calculations for the materials studied.

The lower surface film transmittance coefficient used in the mathematical analysis of heat flow was determined from the measured heat flow and the temperatures of the inner surface and the air 6 in. below the surface. Since the lower surfaces of all panels exposed to the air were painted with lampblack and shellac, the same coefficient applies to all. Analysis of many results for the

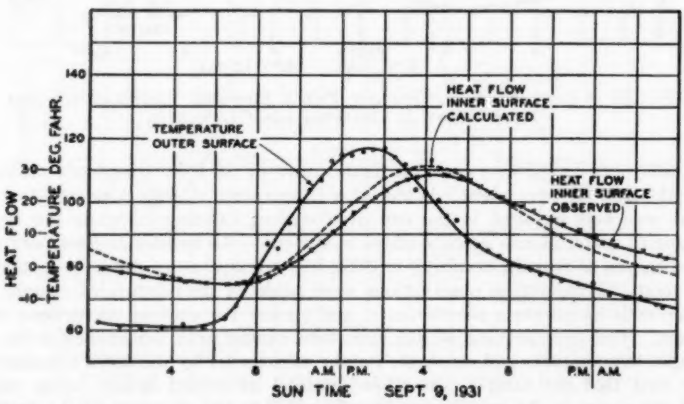


FIG. 25. CURVES FOR 6-IN. CONCRETE PANEL SHOWING OUTER SURFACE TEMPERATURE, OBSERVED INNER SURFACE HEAT FLOW, AND INNER SURFACE HEAT FLOW AS CALCULATED BY THE EMPIRICAL SOLUTION

lower surface gave 1.9 Btu per hour per square foot per degree temperature difference, which value is used in this paper. Many other data resulting from the study are available for analysis of the film transmittance for the lower surface and for the upper surface with still and moving air as measured at night, on clear and cloudy days, and for other conditions.

MATHEMATICAL ANALYSIS OF PERIODIC HEAT FLOW THROUGH ROOF

It is desired to develop as simple a method as possible for use in determining the rate of heat flow into an enclosure, using outside weather conditions and the physical properties of the structure. Such method is available for steady

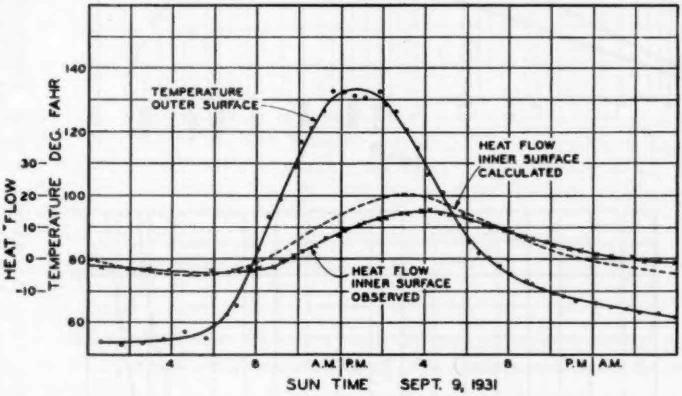


FIG. 26. CURVES FOR 4-IN. GYPSUM PANEL SHOWING OUTER SURFACE TEMPERATURE, OBSERVED INNER SURFACE HEAT FLOW, AND INNER SURFACE HEAT FLOW AS CALCULATED BY THE EMPIRICAL SOLUTION

state conditions where the outside and inside air or surface temperatures have remained constant for a sufficient length of time for the rate of heat flow to become constant. It appears, however, that there is no method available for the case considered in this paper, where the outside source of heat and the outside surface temperature vary as a harmonic function of time.

It has already been pointed out that the problem of starting with the outside source of heat from the sun and the outside air temperature is infinitely more complicated than the problem of starting from the outside surface temperature; this paper is limited to a solution of the latter part of the complicated problem, leaving for future analysis the problem of arriving at the outside surface temperature using as factors the intensity of solar radiation, the outside weather conditions, and the physical properties of the structure. Stated more specifically, the problem is to take the outside surface temperature curve for a 24-hr period, as plotted in any of the Figs. 9 to 26, the thickness, conductivity, density and specific heat of the panel, the inside air temperature, and the inside surface film resistance; and from these to determine by calculation the rate at any

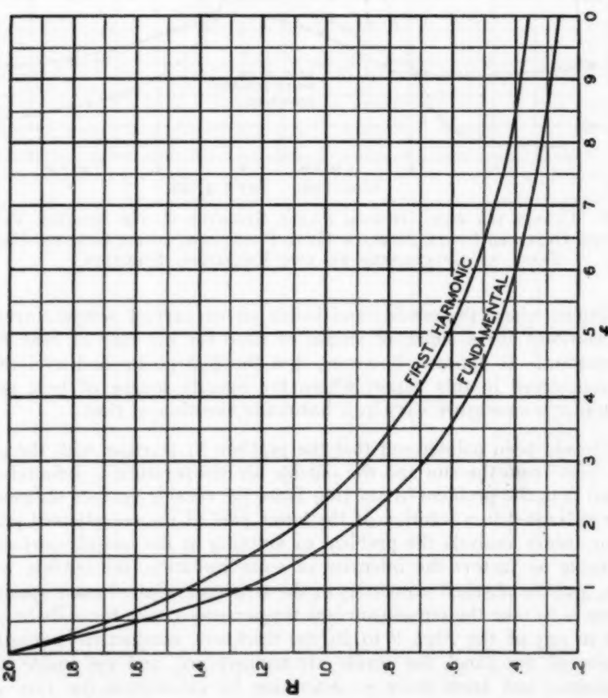


FIG. 27. CURVES GIVING R FOR THE FUNDAMENTAL AND FIRST HARMONIC WAVES USED IN CALCULATING THE RATIO OF THE AMPLITUDE OF THE TEMPERATURE OSCILLATION ON THE OUTSIDE WALL SURFACE TO THAT ON THE INSIDE

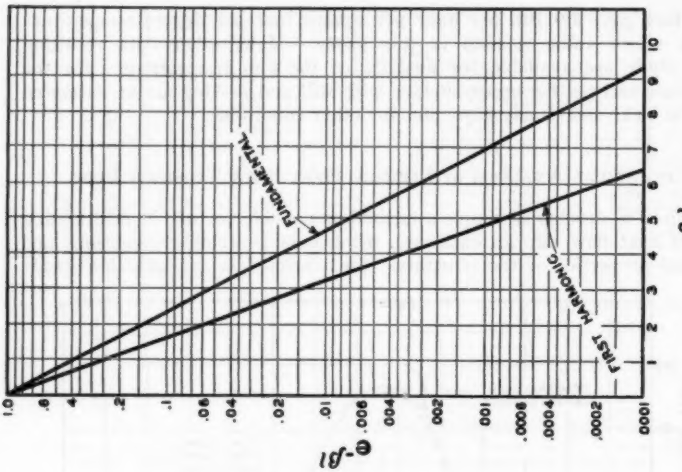


FIG. 28. CURVES GIVING $e^{-\beta l}$ FOR THE FUNDAMENTAL AND FIRST HARMONIC WAVES USED IN CALCULATING THE RATIO OF THE AMPLITUDE OF THE TEMPERATURE OSCILLATION OF THE OUTSIDE WALL SURFACE TO THAT ON THE INSIDE

instant of heat flow from the inside surface to the air. The nearest approach to a solution of this has been in the work of Ingersoll and Zobel,[7] concerning the rate of heat transfer at any point within an infinitely thick solid, resulting from the application of simple harmonic temperature variation at the outside surface. While the Ingersoll and Zobel solution serves the problem in hand better than the assumption of steady state conditions, it is not adaptable to the needs of the air-conditioning engineer.

Mathematically, the solution of the following equation of heat flow is desired:

$$\frac{\partial \theta}{\partial t} = h^2 \frac{\partial^2 \theta}{\partial x^2} \tag{2}$$

which will satisfy the boundary conditions:

$$\theta = \theta_0 \sin \omega t \text{ at } x = 0, \text{ and}$$

$$k \frac{\partial \theta}{\partial x} + \epsilon \theta = 0 \quad \text{at } x = l \tag{4}$$

For the sake of simplicity, it is assumed that the average temperature over a complete cycle is zero on both sides of the wall. No generality is lost by this assumption, for when the average temperatures on either side of the wall differ from zero, the steady flow of heat due to these average temperatures may be calculated and the two solutions added.

For solving the oscillatory case, the terms are defined as follows:

k = coefficient of thermal conductivity of slab
ρ = density of slab
c = specific heat of slab
ϵ = ratio of the heat flow per unit area from the slab surface to the air, to the difference in temperature between this surface and air. Assumed constant.
l = thickness of the slab

$\omega = \frac{2\pi}{T}$ where T is the period of the temperature oscillations

$$h = \sqrt{\frac{k}{c\rho}}$$

$$\beta = \frac{1}{h}\sqrt{\frac{\omega}{2}} = \sqrt{\frac{c\rho\omega}{2k}}$$

$$f = \frac{\epsilon}{k\beta} = \frac{\epsilon\sqrt{2}}{\sqrt{c\rho\omega k}}$$

The solution is

$$\theta = c_1 e^{-\beta x} \sin (\omega t + \gamma_1 - \beta x) + c_2 e^{\beta(x - 2l)} \sin [\omega t + \gamma_2 + \beta (x - 2l)] \tag{5}$$

The first term in Equation 5 represents the direct wave and the second term, a wave reflected from the inner surface. In the general case, there may be multiple reflections as in the optical case, and so each term must represent the

[7] See Bibliography, c.

sum of an infinite series of waves. However, where dealing with insulating materials, the amplitude of the temperature wave is so rapidly damped out that only the first reflection from the inner surface is of importance. Equation 5 may be written:

$$\theta = \frac{\theta_o}{B} e^{-\beta x} \sin (\omega t + \alpha - \beta x)$$

$$+ \frac{A\theta_o}{B} e^{\beta (x - 2l)} \sin [\omega t + \alpha + \sigma + \beta (x - 2l)] \tag{6}$$

where

$$A = + \sqrt{\frac{(1-f)^2+1}{(1+f)^2+1}}$$

$$\sigma = \cos^{-1} \frac{2-f^2}{\sqrt{4+f^4}} \qquad 0 \leqq \sigma \leqq \pi$$

$$\alpha = \sin^{-1} \frac{A}{B} e^{-2\beta l} \sin (\sigma - 2\beta l) \qquad (-\pi < \alpha < \pi)$$

$$B = \sqrt{1 + A^2 e^{-4\beta l} + 2Ae^{-2\beta l} \cos (\sigma - 2\beta l)} \tag{7}$$

In practice, $e^{-2\beta l}$ will be small, and $e^{-4\beta l}$ will be very small, while A must lie between 0.546 and $+1.00$. Therefore, $B = 1 + Ae^{-2\beta l} \cos (\sigma - 2\beta l)$ approximately, which shows that B may be set equal to 1, and α equal to zero. The resulting solution

$$\frac{\theta}{\theta_o} = e^{-\beta x} \sin (\omega t - \beta x) + Ae^{\beta (x - 2l)} \sin [\omega t + \sigma + \beta (x - 2l)]$$

will not be in error by more than $e^{-2\beta l}$ in unity.

The problem's main interest lies in the solution of the temperature and heat flow from the inner surface of the wall where $x = l$

Here $\frac{\theta}{\theta_o} = \frac{\theta_1}{\theta_o} = e^{-\beta l} [\sin (\omega t - \beta l) + A \sin (\omega t - \beta l + \sigma)]$

which obviously represents a simple sine function of amplitude

$$\frac{\theta_m}{\theta_o} = Re^{-\beta l} \textit{ where} \tag{8}$$

$$R = + \sqrt{1 + A^2 + 2A \cos \sigma}$$

$\frac{\theta_m}{\theta_o}$ is the ratio of the amplitudes of the temperature variations on the inside of the wall to that on the outside. Since A and σ are functions of the single variable f, R is also a function of f, and is plotted in Fig. 27 for both the fundamental and first harmonic waves. In a like manner, values of $e^{-\beta l}$ are potted against βl in Fig. 28.

To find the value of $\frac{\theta_m}{\theta_o}$ it is only necessary to calculate f and βl from the constants of the slab, and take the product $Re^{-\beta l}$ from Fig. 27 and Fig. 28. The temperature at the inner surface may now be expressed by the equation

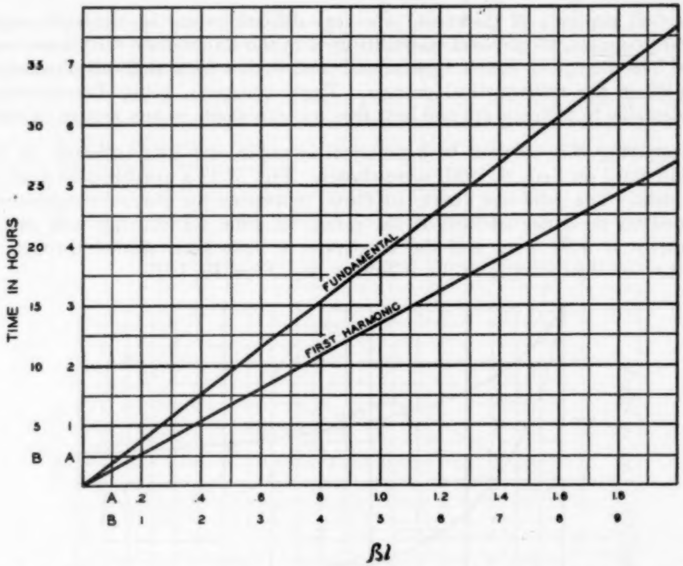


FIG. 29. CURVES GIVING UNCORRECTED TIME IN HOURS REQUIRED FOR THE CREST OF THE FUNDAMENTAL AND FIRST HARMONIC WAVES TO PASS THROUGH A PANEL

$$\theta_1 = \theta_m \sin \omega \; (t + t_o) \tag{9}$$

where

t_o = the time between the occurrences of the temperature maxima at the outside and inside surfaces. The temperature is maximum at the outside surface when $\omega t = \frac{\pi}{2}$, and the corresponding maximum at the inside surface occurs when

$$\tan\,(\omega t - \beta l) = \frac{1 + A\cos\sigma}{A\sin\sigma} \textit{ where } (\omega t - \beta l) \text{ is in the first quadrant.}$$

$$\text{Then } \omega t_o = \left\{ \tan^{-1}\left(\frac{1 + A\cos\sigma}{A\sin\sigma}\right) - \frac{\pi}{2} \right\} + \beta l \tag{10}$$

After f and βl have been calculated, curves may be drawn for rapid solution of the timing of the waves. Fig. 29 gives the uncorrected time in hours required for the crest of the fundamental and first harmonic waves to pass through a panel. Fig. 30 gives the correction time in hours to be applied in determining the correct length of time required for the fundamental and first harmonic waves to pass through a panel.

Because the English system of units combines terms, which because of their dimensions (*i.e.* feet with inches in Btu per square foot per hour per degree

Fahrenheit per inch of thickness) are very difficult to use in involved mathematical processes, all physical constants used in the Laboratory work were converted into the cgs or metric system, and used in that form until all dimensions cancelled in the mathematical process. These constants, being dimensionless, are adaptable to temperature and heat flow measurements in any system of units

In applying this solution to a practical example, the data collected on the gypsum panel on Sept. 9, 1931, were chosen. Fig. 31 is a graphical analysis of these data. The solid line curve, *BCDEG,* represents the plot of temperatures obtained on the outer surface of the panel. A time for starting was chosen arbitrarily at 4:00 A.M., and the temperatures have been plotted through to 4:00 A.M. on the morning of the following day, Sept. 10, 1931.

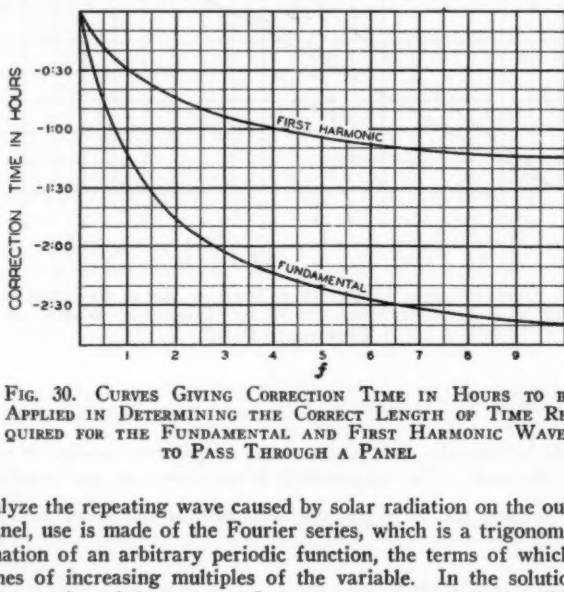


FIG. 30. CURVES GIVING CORRECTION TIME IN HOURS TO BE APPLIED IN DETERMINING THE CORRECT LENGTH OF TIME REQUIRED FOR THE FUNDAMENTAL AND FIRST HARMONIC WAVES TO PASS THROUGH A PANEL

To analyze the repeating wave caused by solar radiation on the outer surface of the panel, use is made of the Fourier series, which is a trigonometric series approximation of an arbitrary periodic function, the terms of which are sines and cosines of increasing multiples of the variable. In the solution, the approximate equation of the outer surface temperature curve is found by analysis of the curve. A Fourier series consists of an infinite number of terms; but for ordinary heat flow work a summation of only the first two terms, namely the fundamental and the first harmonic, is sufficient. To determine the constants entering into the series, the temperature curve for the outer surface may be analyzed by the use of either a polar planimeter or a harmonic analyzer, or by computing mathematical averages, which last-named method was used at the Laboratory because it required no special equipment. The 2-term mathematical series representing the outer surface temperature was first computed; from this the amplitudes of the fundamental and first harmonic waves at the outer surface were obtained. By applying the theoretical formulae to these amplitudes of the outer surface, the corresponding fundamental and first harmonic amplitudes of the inner surface were obtained. These inner surface

amplitudes were vectorially added to give the wave heat flow, which was added algebraically to the heat flow in the panel (considered to be steady) to give the heat flow at the inner surface.

1. On the Fourier Analysis Chart, Table 5, the starting time is filled in at the top of the sheet. The time starts at 4:00 A.M. and continues down the sheet at 20-min increments until 3:40 the following morning has been reached. Thus 72 times have been listed.

2. On each line of the Fourier Analysis Chart, or every twenty minutes during the 24-hr period, the value of the surface temperature above zero

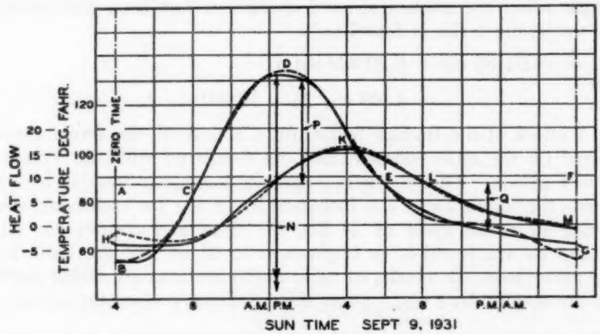


FIG. 31. MATHEMATICAL ANALYSIS OF HEAT FLOW THROUGH 4-IN. GYPSUM PANEL

Fahrenheit is recorded in column *a*. Thus, on Fig. 31 at *N*, which corresponds to 12:20 P.M., the value of the curve is 132.3 F.

3. On the chart, Table 5, the 72 readings in column *a* total 6273.90 F which when divided by 72 gives 87.14 F as the value of the average integrated temperature. This temperature line, *AF*, will divide the curve, Fig. 31, so that the sum of the solid line areas *ABC* and *EFG* will be exactly equal to the area *CDE*. In other words, over a 24-hr period, the area above the average integrated temperature line will be exactly equal to the area below it.

4. The value of the average integrated temperature is algebraically subtracted from the temperature listed in column *a* to obtain the values listed in column *b*. These values show the height of the curve *BCDEG* above or below the average integrated temperature line, 87.14 F. Thus, at 1:40 P.M. there is a height of 43.96 F as shown by *P*, while at 11:20 P.M. the height is —19.84 F as shown by *Q*.

5. The values in column *b* are multiplied by the listed values of cos *a*, sin *a*, cos 2*a* and sin 2*a* to obtain the respective columns *b* cos *a*, *b* sin *a*, *b* cos 2*a*, and *b* sin 2*a*. All changes in sign in the cosine and sine are taken into account.

6. The 72 readings in each of the columns mentioned in Paragraph 5 are algebraically added. Each of these 4 totals is divided by 36 to obtain the coefficients for the first two terms, the fundamental and first harmonic of the Fourier series. The series is written:

$$\theta_{\substack{\text{outer}\\\text{surface}}} = A \cos a + B \sin a + C \cos 2a + D \sin 2a + + + \tag{11}$$

where

A is the coefficient determined from the $b \cos a$ column

B is the coefficient determined from the $b \sin a$ column

C is the coefficient determined from the $b \cos 2a$ column

D is the coefficient determined from the $b \sin 2a$ column

In Equation 11, the terms containing A and B will give the fundamental of the series, while the terms containing C and D give the first harmonic. In the example when the aforementioned operations had been performed on the algebraic totals, the series is found to be:

$$\theta_{\substack{\text{outer}\\\text{surface}}} = -27.186 \cos a + 20.929 \sin a$$
$$-4.308 \cos 2a - 12.900 \sin 2a + + + \tag{12}$$

6a. If a check of the surface temperature is desired, the actual temperature, as measured on the outer surface, may be compared with the Fourier series over a 24-hr period. This check is shown in Table 6. Starting at 4:00 A.M., or zero time, the timing angle a is increased by 15 deg for each hour, since there are 360 deg in a full cycle of 24 hr; the sines and cosines for all of the angles a and $2a$ are found from trigonometric tables, and the indicated multiplications performed; the results of these multiplications are added algebraically to obtain the amplitude, $\theta_{\substack{\text{outer}\\\text{surface}}}$; and this resulting temperature wave is added to the value of the average integrated temperature to obtain a temperature which will approximate the experimental temperature curve at the outer surface. The resulting broken line curve, *BCDEG*, in Fig 31 is the check on the accuracy of the 2-term Fourier analysis.

7. From the English or engineering values of the physical constants of the wall, namely

$k = 1.445$ Btu per square foot per hour per degree Fahrenheit per inch

$\rho = 64.89$ pounds per cubic foot

$c = 0.234$

$h_1 = 1.9$ Btu per square foot per hour per degree Fahrenheit

$l = 4.188$ in.

which have been determined previously by test and measurement, the equivalent cgs values are found to be

$k = 1.445 \times 3.44 \times 10^{-4} = 4.971 \times 10^{-4}$ cal per square centimeter per centimeter per second per degree centigrade

$\rho = 64.89 \div 62.4 = 1.04$ grams per cubic centimeter

$c = 0.234$

$h_1 = 1.9 \times 1.355 \times 10^{-4} = 2.574 \times 10^{-4}$ cal per square centimeter per second per degree centigrade

$l = 4.188 \times 2.54 = 10.637$ centimeters

8. The value of β is determined from the formula

$$\beta = \sqrt{\frac{c\rho\omega}{2k}} \tag{13}$$

where

$$\omega = \frac{2\pi}{T}$$

T = the length of the period. In the cgs system, $T = 86{,}400$ seconds for a 24-hr period.

In the example

$$\beta = \sqrt{\frac{0.234 \times 1.04 \times 3.1416}{86{,}400 \times 4.971 \times 10^{-4}}} = 0.1334$$

9. The value of E is determined from the formula

$$E = \frac{h_1}{k}$$

and in the example

$$E = \frac{2.574 \times 10^{-4}}{4.971 \times 10^{-4}} = 0.5178 \tag{14}$$

10. The value of f is determined from the formula

$$f = \frac{E}{\beta} \tag{15}$$

and in the example

$$f = \frac{0.5178}{0.1334} = 3.881$$

11. The value of βl is determined from the formula

$$\beta l = \beta \times l \tag{16}$$

and in the example

$$\beta l = 0.1334 \times 10.637 = 1.419$$

12. Next, the value of the fundamental amplitude at the outer surface and the time in hours that its crest is distant from zero time are found by analyzing the fundamental—or A and B—terms of the Fourier equation of the wave form at the outer surface, by drawing the vector diagram as shown in A in Fig. 32. Here, A is laid off on the y axis taking into account its sign. In the example, -27.186 is the A term, and this is laid off on the y axis in a downward direction from the origin. The B term is laid off on the x axis also taking into account its sign. In the example, 20.929 is the β term, and this is laid off on the x axis to the right. The resultant, which is the value of the fundamental amplitude at the outer surface, is therefore

$$\theta_{\substack{\text{fundamental}\\ \text{outer surface}}} = \sqrt{27.186^2 + 20.929^2} = 34.30$$

and the timing is the angular distance from the positive y axis measured in a clockwise direction, or

$$90 \text{ deg} + \text{arc tan}\,\frac{27.186}{20.929} = 142 \text{ deg } 26 \text{ min.}$$

Expressed in hours, this is 9 hr 30 min, and is the time it takes the crest of the fundamental wave to travel from zero time to the time of maximum crest at

TABLE 5. FOURIER ANALYSIS CHART FOR OUTER SURFACE TEMPERATURE

GYPSUM PANEL SEPT. 9, 1931

ZERO TIME 4 A.M. AV. TEMP. 87.14 F.

Time	a	b	α	Cos α	b Cos α	Sin α	b Sin α	2α	Cos 2α	b Cos 2α	Sin 2α	b Sin 2α
4:00	54.8	-32.34	0	1.000	-32.34	0000	.00	0	1.000	-32.34	0000	.00
20	54.8	-32.34	5	.996	-32.21	.087	-2.81	10	.985	-31.86	.174	-5.63
40	54.9	-32.24	10	.985	-31.76	.174	-5.61	20	.940	-30.31	.342	-11.03
5:00	54.9	-32.24	15	.966	-31.14	.259	-8.35	30	.866	-27.92	.500	-16.12
20	55.0	-32.14	20	.940	-30.21	.342	-10.99	40	.766	-24.62	.642	-20.63
40	56.0	-31.14	25	.906	-28.21	.423	-13.17	50	.642	-19.99	.766	-23.86
6:00	58.0	-29.14	30	.866	-25.24	.500	-14.57	60	.500	-14.57	.866	-25.24
20	60.6	-26.54	35	.819	-21.74	.574	-15.23	70	.342	-9.08	.940	-24.95
40	64.0	-23.14	40	.766	-17.73	.642	-14.86	80	.174	-4.03	.985	-22.79
7:00	67.4	-19.74	45	.707	-13.96	.707	-13.96	90	0000	.00	1.000	-19.74
20	72.0	-15.14	50	.642	-9.72	.766	-11.60	100	-.174	2.63	.985	-14.91
40	77.2	-9.94	55	.574	-5.71	.819	-8.14	110	-.342	3.40	.940	-9.34
8:00	82.4	-4.74	60	.500	-2.37	.866	-4.10	120	-.500	2.37	.866	-4.10
20	87.7	0.56	65	.423	0.24	.906	.51	130	-.642	-.36	.766	0.43
40	93.3	6.16	70	.342	2.11	.940	5.79	140	-.766	-4.72	.642	3.96
9:00	98.8	11.66	75	.259	3.02	.966	11.26	150	-.866	-10.10	.500	5.83
20	104.0	16.86	80	.174	2.93	.985	16.61	160	-.940	-15.85	.342	5.77
40	109.5	22.36	85	.087	1.95	.996	22.27	170	-.985	-22.02	.174	3.89
10:00	114.6	27.46	90	0000	.00	1.000	27.46	180	-1.000	-27.46	0000	.00
20	119.0	31.86	95	-.087	-2.77	.996	31.73	190	-.985	-31.38	-.174	-5.54
40	123.0	35.86	100	-.174	-6.24	.985	35.32	200	-.940	-33.71	-.342	-12.26
11:00	126.1	38.96	105	-.259	-10.09	.966	37.64	210	-.866	-33.74	-.500	-19.48
20	128.8	41.66	110	-.342	-14.26	.940	39.16	220	-.766	-31.91	-.642	-26.75
40	130.7	43.56	115	-.423	-18.43	.906	39.47	230	-.642	-27.97	-.766	-33.37
12:00	131.8	44.66	120	-.500	-22.33	.866	38.68	240	-.500	-22.33	-.866	-38.63
20	132.3	45.16	125	-.574	-25.92	.819	36.99	250	-.342	-15.44	-.940	-42.45
40	132.4	45.26	130	-.642	-29.06	.766	34.67	260	-.174	-7.88	-.985	-44.58
1:00	132.2	45.06	135	-.707	-31.86	.707	31.86	270	0000	.00	-1.000	-45.06
20	131.9	44.76	140	-.766	-34.29	.642	28.74	280	.174	7.79	-.985	-44.09
40	131.1	43.96	145	-.819	-36.00	.574	25.23	290	.342	15.03	-.940	-41.32
2:00	129.9	42.76	150	-.866	-37.03	.500	21.38	300	.500	21.38	-.866	-37.03
20	128.0	40.86	155	-.906	-37.02	.423	17.28	310	.642	26.23	-.766	-31.30
40	125.6	38.46	160	-.940	-36.15	.342	13.15	320	.766	29.46	-.642	-24.69
3:00	122.3	35.16	165	-.966	-33.96	.259	9.12	330	.866	30.45	-.500	-17.58
20	118.5	31.36	170	-.985	-30.89	.174	5.46	340	.940	29.48	-.342	-10.73
40	114.0	26.86	175	-.996	-26.75	.087	2.34	350	.985	26.46	-.174	-4.67
4:00	109.4	22.26	180	-1.000	-22.26	0000	.00	360	1.000	22.26	0000	.00
20	104.3	17.16	185	-.996	-17.09	-.087	-1.49	10	.985	16.90	.174	2.99
40	100.0	12.86	190	-.985	-12.67	-.174	-2.24	20	.940	12.09	.342	4.40
5:00	96.4	9.26	195	-.966	-8.95	-.259	-2.40	30	.866	8.02	.500	4.63
20	92.9	5.76	200	-.940	-5.41	-.342	-1.97	40	.766	4.41	.642	3.70
40	89.5	2.36	205	-.906	-2.14	-.423	-1.00	50	.642	1.52	.766	1.81
6:00	86.7	-0.44	210	-.866	0.38	-.500	.22	60	.500	-0.22	.866	-0.38
20	84.2	-2.94	215	-.819	2.41	-.574	1.69	70	.342	-1.01	.940	-2.76
40	82.0	-5.14	220	-.766	3.94	-.642	3.30	80	.174	-0.89	.985	-5.06
7:00	80.3	-6.84	225	-.707	4.84	-.707	4.84	90	0000	.00	1.000	-6.84
20	79.0	-8.14	230	-.642	5.23	-.766	6.24	100	-.174	1.42	.985	-8.02
40	77.4	-9.74	235	-.574	5.59	-.819	7.98	110	-.342	3.33	.940	-9.16
8:00	75.9	-11.24	240	-.500	5.62	-.866	9.73	120	-.500	5.62	.866	-9.73
20	74.3	-12.84	245	-.423	5.43	-.906	11.63	130	-.642	8.24	.766	-9.84
40	73.0	-14.14	250	-.342	4.84	-.940	13.29	140	-.766	10.83	.642	-9.08
9:00	72.2	-14.94	255	-.259	3.87	-.966	14.43	150	-.866	12.94	.500	-7.47
20	71.3	-15.84	260	-.174	2.76	-.985	15.60	160	-.940	14.89	.342	-5.42
40	70.5	-16.64	265	-.087	1.45	-.996	16.57	170	-.985	16.39	.174	-2.90
10:00	69.8	-17.34	270	0000	.00	-1.000	17.34	180	-1.000	17.34	0000	.00
20	69.0	-18.14	275	.087	-1.58	-.996	18.07	190	-.985	17.87	-.174	3.16
40	68.3	-18.84	280	.174	-3.28	-.985	18.56	200	-.940	17.71	-.342	6.44
11:00	67.9	-19.24	285	.259	-4.98	-.965	18.69	210	-.866	16.66	-.500	9.62
20	67.3	-19.84	290	.342	-6.79	-.940	18.65	220	-.766	15.20	-.642	12.74
40	66.8	-20.34	295	.423	-8.60	-.906	18.43	230	-.642	13.06	-.766	15.58
12:00	66.2	-20.94	300	.500	-10.47	-.866	18.15	240	-.500	10.47	-.866	18.13
20	65.9	-21.24	305	.574	-12.19	-.819	17.40	250	-.342	7.26	-.940	19.97
40	65.3	-21.84	310	.642	-14.02	-.766	16.60	260	-.174	3.80	-.985	21.51
1:00	64.9	-22.24	315	.707	-15.72	-.707	15.72	270	0000	.00	-1.000	22.24
20	64.4	-22.74	320	.766	-17.42	-.642	14.60	280	.174	-3.96	-.985	22.40
40	64.0	-23.14	325	.819	-18.95	-.574	13.28	290	.342	-7.91	-.940	21.75
2:00	63.6	-23.54	330	.866	-20.39	-.500	11.77	300	.500	-11.77	-.866	20.39
20	63.2	-23.94	335	.906	-21.69	-.423	10.13	310	.642	-15.37	-.766	18.34
40	63.0	-24.14	340	.940	-22.69	-.342	8.26	320	.766	-18.49	-.642	15.50
3:00	62.8	-24.34	345	.966	-23.51	-.259	6.30	330	.866	-21.08	-.500	12.17
20	62.5	-24.64	350	.985	-24.27	-.174	4.29	340	.940	-23.16	-.342	8.43
40	62.2	-24.94	355	.996	-24.84	-.087	2.17	350	.985	-24.57	-.174	4.34
Totals	6273.90				-978.68		+753.44			-155.10		-464.41
Average	87.14				A = -2.7186		B = +2.0929			C = -.4308		D = -1.2900

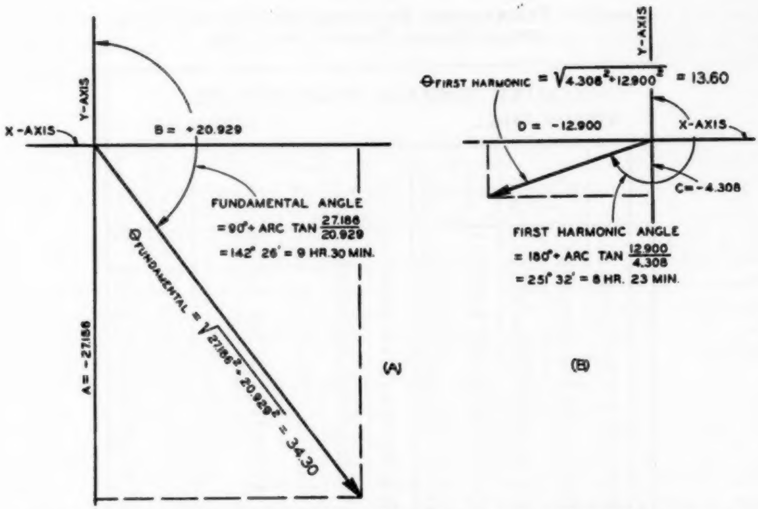


FIG. 32. VECTORIAL DETERMINATION OF FUNDAMENTAL AND FIRST HARMONIC AMPLITUDE AT OUTER SURFACE

the outer surface. This is found by dividing the angular distance by 15 deg. the number of degrees in 1 hr of the fundamental.

13. The value of the first harmonic amplitude at the outer surface is next determined, as is also the time in hours that its crest is distant from zero time. These are found by vector analysis exactly as were those of the previous determination. As shown in *B* of Fig. 32, the constant in the Fourier equation of the wave form at the outer surface *C*, or —4.308, is laid off on the *y* axis in a downward direction. The constant *D*, —12.900, in the same equation, is laid off on the *x* axis to the left. The resultant is the value of the amplitude of the first harmonic at the outer surface

$$\theta_{\substack{\text{first harmonic}\\ \text{outer surface}}} = \sqrt{4.308^2 + 12.900^2} = 13.60$$

Again, the timing is the angular distance from the *y* axis measured in a clockwise direction, or

$$180 \text{ deg} + \text{arc tan} \frac{12.900}{4.308} = 251 \text{ deg } 32 \text{ min}$$

Expressed in hours, it takes the crest of the first harmonic 8 hr 23 min, to travel from zero time to the time of maximum crest at the outer surface, since for the harmonic, there are 30 deg in 1 hr.

14. From the value of *f* and the curves in Fig. 27, *R* is determined to be 0.560 for the fundamental and 0.740 for the first harmonic wave.

15. From the value of βl and the curves in Fig. 28, $e^{-\beta l}$ is determined to be 0.241 for the fundamental and 0.137 for the first harmonic wave.

TABLE 6. TEMPERATURES RESULTING FROM FOURIER ANALYSIS OF OUTSIDE TEMPERATURE CURVE

OUTER SURFACE TEMPERATURE

GYPSUM PANEL — SEPT. 9, 1931

Time	(a) α	(b) Cos α	(c) (b)×A	(d) Sin α	(e) (d)×B	(f) 2α	(g) Cos 2α	(h) (g)×C	(j) Sin 2α	(k) (j)×D	(m) (c+e+h+k)	(n) (m)+Av Temp
4	0	1000	-2719	000	.00	0	1000	-431	.000	00	-3150	5564
5	15	.966	-2626	.259	542	30	.866	-373	.500	- 645	-3102	5612
6	30	866	-2354	.500	1046	60	.500	-215	866	-1117	-2640	6074
7	45	707	-1922	.707	1480	90	000	00	1.000	-1290	-17.32	6982
8	60	.500	-1359	.866	1812	120	- 500	215	866	-11.17	- 449	8265
9	75	259	- 704	.966	2021	150	- .866	373	500	- 6.45	1045	9759
10	90	000	.00	1000	2093	180	-1.000	431	000	00	2524	11238
11	105	- .259	704	.966	2021	210	- .866	373	- 500	645	3743	12457
12	120	- 500	13.59	.866	1812	240	- .500	215	- 866	1117	4503	132.17
1	135	- 707	1922	.707	1480	270	.000	.00	-1000	1290	46.92	13406
2	150	- 866	2354	.500	1046	300	.500	-215	- 866	1117	4302	13016
3	165	- 966	2626	.259	542	330	.866	-373	- 500	645	3440	12154
4	180	-1.000	2719	.000	.00	0	1.000	-431	000	00	2288	110.02
5	195	- .966	2626	-.259	- 542	30	.866	-373	500	- 645	1066	9780
6	210	- .866	2354	-.500	-1046	60	.500	-215	866	-11.17	- .24	8690
7	225	- 707	1922	-.707	-1480	90	.000	.00	1000	-1290	- 848	7866
8	240	- .500	1359	-.866	-18.12	120	- .500	215	866	-1117	-1355	7359
9	255	- .259	7.04	-.966	-2021	150	- .866	373	500	- 645	-1589	7125
10	270	.000	00	-1000	-2093	180	-1.000	431	000	00	-1662	7062
11	285	.259	- 704	-.966	-20.21	210	- .866	373	- 500	645	-1707	7007
12	300	.500	-1359	-.866	-18.12	240	- .500	215	- 866	11.17	-1839	6875
1	315	.707	-1922	-.707	-1480	270	.000	.00	-1000	1290	-21.12	6602
2	330	.866	-2354	-.500	-1046	300	.500	-215	- 866	1117	-2498	6216
3	345	.966	-2626	-.259	- 542	330	.866	-373	- 500	645	-2896	5818

TABLE 7. CALCULATED HEAT FLOWS THROUGH INNER SURFACE

HEAT FLOW ANALYSIS

GYPSUM PANEL — SEPT 9, 1931

Time	(a) Fund Angle	(b) Sin a	(C) (b)×θ₁ 1st Har Inner	(d) 1st Har Angle	(e) Sin d	(f) (e)×θ₂ 1st Har Inner	(g) (C)+(f)	(j) (g)×h_i	(k) (j)+steady Flow
4	259°15'	-.982	-4.55	133°0'	.920	1.27	-3.28	-6.23	-1.10
5	274°-	-.997	-4.61	143°-	.620	.83	-3.78	-7.18	-2.05
6	289°-	-.944	-4.37	173°-	.122	.17	-4.20	-7.98	-2.85
7	304°-	-.827	-3.83	203°-	-.391	-.54	-4.37	-8.30	-3.17
8	319°-	-.653	-3.02	233°-	-.798	-1.10	-4.12	-7.83	-2.70
9	334°-	-.434	-2.00	263°-	-.993	-1.37	-3.37	-6.40	-1.27
10	349°-	-.186	-.86	293°-	-.920	-1.27	-2.13	-4.04	1.09
11	4°15'	.074	.34	323°-	-.602	-.83	-.49	-.93	4.20
12	19°-	.330	1.53	353°-	-.122	-.17	1.36	2.58	7.71
1	34°-	.563	2.60	23°0'	.391	.54	3.14	5.97	11.10
2	49°-	.757	3.50	53°-	.798	1.10	4.60	8.74	13.87
3	64°-	.901	4.17	83°-	.993	1.37	5.54	10.52	15.65
4	79°-	.982	4.55	113°-	.920	1.27	5.82	11.06	16.19
5	94°-	.997	4.61	143°-	.602	.83	5.44	10.34	15.47
6	109°-	.944	4.37	173°-	.122	.17	4.54	8.63	13.76
7	124°-	.827	3.83	203°-	-.391	-.54	3.29	6.26	11.38
8	139°-	.653	3.02	233°-	-.798	-1.10	1.92	3.65	8.78
9	154°-	.434	2.00	263°-	-.993	-1.37	.63	1.19	6.32
10	169°-	.186	.86	293°-	-.920	-1.27	-.41	-.78	4.35
11	184°-	-.074	-.34	323°-	-.602	-.83	-1.17	-2.22	2.91
12	199°-	-.330	-1.53	353°-	-.122	-.17	-1.70	-3.23	1.90
1	214°-	-.563	-2.60	23°-	.391	.54	-2.06	-3.91	1.22
2	229°-	-.757	-3.50	53°-	.798	1.10	-2.40	-4.66	.57
3	244°-	-.901	-4.17	83°-	.993	1.37	-2.80	-5.32	-.19

16. The value of the amplitude of the fundamental at the inner surface is determined from the multiplication of the amplitude of the fundamental at the outer surface by the factors R and $e^{-\beta l}$ for the fundamental. In the example,

$$\theta_{\substack{\text{fundamental}\\ \text{inner surface}}} = 34.30 \times 0.560 \times 0.241 = 4.629$$

17. The value of the amplitude of the first harmonic at the inner surface is determined from the multiplication of the amplitude of the first harmonic at the outer surface by the factors R and $e^{-\beta l}$ for the first harmonic. In the example,

$$\theta_{\substack{\text{first harmonic}\\ \text{inner surface}}} = 13.60 \times 0.740 \times 0.137 = 1.379.$$

18. Next, the uncorrected time in hours and minutes it takes the crest of the fundamental wave to travel through the panel is determined. On the fundamental curve in Fig. 29, the time is read from the value of βl to be 5 hr 25 min. From this is subtracted the corrective time, 2 hr 12 min read from f on the fundamental curve in Fig. 30. This 3 hr 13 min is the true time it takes the crest of the fundamental wave to pass through the finite wall.

19. The uncorrected time in hours and minutes it takes the crest of the first harmonic wave to travel through the panel is determined. On the first harmonic curve in Fig. 29, the time is read from the value of βl to be 3 hr 50 min. From this is subtracted the corrective time, 59 min, read from f on the first harmonic curve in Fig. 30. This 2 hr 51 min is the true time it takes the crest of the first harmonic wave to pass from the outside to the inside of the finite wall.

20. The time the crest of the fundamental wave is distant from 12 midnight of the night previous is determined by adding together the time from midnight to zero time, the time from zero time to the time of maximum crest of the fundamental, and the corrected time it takes the wave to pass through the wall. This is done in the example:

4 hr	00 min
9 hr	30 min
3 hr	13 min
16 hr	43 min

Therefore, at 4:43 P.M. there is a crest of the fundamental at the inner surface.

21. The time the crest of the first harmonic wave is distant from 12 midnight of the night previous is determined by adding together the time from midnight to zero time, the time from zero time to the time of maximum crest of the first harmonic, and the corrected time it takes the first harmonic to pass through the wall. This is done in the example:

4 hr	00 min
8 hr	23 min
2 hr	51 min
15 hr	14 min

Therefore, at 3:14 P.M. there is a crest of the first harmonic at the inner surface.

22. For ease of computation, it is desirable to determine, from the time of the crest of the fundamental wave at the inner surface, when the timing angle of the fundamental wave will be equal to zero deg. To obtain this time, it is only necessary to subtract 6 hr from the time of the crest—the equivalent of subtracting 90 deg from the timing angle. Since 15 deg are equal to 1 hr for the fundamental wave, the angle of the fundamental at each hour of the day is easily found by interpolation. In the example:

4:43 P.M. —6 hr = 10:43 A.M. at zero angle.
At 11:00 A.M., the angle of the fundamental

$$= \frac{17 \times 15}{60} = 4 \text{ deg } 15 \text{ min}$$

23. In a manner similar to that described, three hours are subtracted from the time of the crest of the first harmonic wave at the inner surface to obtain the time when the timing angle of the first harmonic wave will be equal to zero deg. Since 30 deg are equal to 1 hr for the first harmonic wave, the angle of the first harmonic at each hour of the day is easily found by interpolation. In the example:

3:14 P.M. —3 hr = 12:14 P.M. at zero angle.
At 1:00 P.M., the angle of the first harmonic

$$= \frac{46 \times 30}{60} = 23 \text{ deg } 0 \text{ min}$$

24. The average integrated temperature at the inner surface over the 24-hr period is determined in the English system of units

$$\frac{(T_1 - T_2)\, k}{l} = (T_2 - T_3)\, h_1 \tag{17}$$

where

T_1 = average integrated outer surface temperature
T_2 = inside surface temperature
T_3 = inside air temperature

In the example:

$$\frac{87.14 - T_2}{4.188} = (T_2 - 69.6)\, 1.9$$

$$(T_2 = 72.30 \text{ F.})$$

25. The steady heat flow through the inner surface is easily determined from the film transmittance coefficient.

$$\text{Btu per square foot per hour} = (T_2 - T_3)\, h_1 \tag{18}$$

and in the example is equal to $(72.30 - 69.6)\, 1.9 = 5.13$ Btu

26. The vector analysis form chart, Table 7, is filled out for the 2-term analysis. The angles of the fundamental waves *a* and first harmonic waves *d* are spaced out, keeping in mind that 15 deg is the equivalent of 1 hr for the fundamental wave and 30 deg is the equivalent of 1 hr for the first harmonic wave. The corresponding sines *b* and *e* are multiplied by $\theta_{\substack{\text{fundamental}\\ \text{inner surface}}}$ or 4.629, and $\theta_{\substack{\text{first harmonic}\\ \text{inner surface}}}$ or 1.379, as shown in *c* and *f* respectively.

27. The items, c and f, are algebraically added to obtain the resulting temperature at the inner surface above and below the average surface temperature as shown in g.

Multiplying these values by the film transmittance coefficient, h_1, gives the amplitude of the heat flow wave in j, and the value of the steady heat flow added to this wave flow results in the calculated heat flow for the wall in k. This calculated curve is sketched in Fig. 31 as the broken line curve, *HJKLM*, which checks the solid line observed heat flow curve *HJKLM*.

The foregoing analysis has been applied to a number of tests made on the panels studied. The inner surface heat flow curves marked *calculated* in Figs. 13 to 23 show how closely the results of the application of the mathematical study fit the measured heat flows through the inner surfaces of the panels.

The calculated heat flow through the lower surface of the pine panel, Fig. 13, fits the observed heat flow almost perfectly in regard to both the amplitude and the time of the crest of the wave. There is usually a small break in the calculated curves at 4 A.M., when the arbitrarily chosen, or zero time, begins and ends the daily cycle. The average integrated temperatures of the outer surface for 2 consecutive days were never quite the same, and since each 24-hr period was calculated as a unit, the adjacent ends of the calculated curves do not abut.

The check between the calculated and observed heat flows through the inner surface of the 6-in. concrete panel, Fig. 14, is not quite as perfect, but considering the greater thickness, conductivity, and heat capacity of this panel, the check is satisfactory. The 4-in. gypsum panel, Fig. 17, gives as close a check as could be desired. The data for Sept. 8 were included for the special purpose of showing the application of the mathematical solution to a day which was far from perfect, and for which the top surface temperature curve deviated greatly from the general type. This day was intermittently and rather densely cloudy from 10 A.M. until 3 P.M. The top surface temperature curve is similar in shape to the observed heat flow curve, and the Fourier analysis of the temperature curve gives a calculated heat flow curve which fits the observed heat flow curve very well.

The checks between the calculated and observed heat flows through the inner surface of the cork-concrete and concrete-cork panels, Figs. 18 and 19, are not as perfect as for the pine and gypsum. The calculated curve for the cork-concrete panel shows about the same amplitude as that for the observed, but its crest is shown to pass through the lower surface about 1½ hr later. The calculated curve for this panel, when the concrete was up, shows a lower amplitude than the observed curve, and lags approximately 3½ hr behind it. The application of the Fourier analysis to the cork-concrete and concrete-cork panels involves the use of the average conductivity, density, and specific heat of the compound panel, so it obviously should not rigorously apply. The curves of Figs. 18 and 19 indicate that the check between observed and calculated heat flow is fair when the insulation is on the outer surface, but unsatisfactory when the insulation is on the inside. These tests indicate that for compound walls, where the physical properties of the component parts differ widely, an adaptation of the Fourier analysis method which considers each component part separately is desirable.

The calculated heat flow does not fit the observed heat flow perfectly for the lower surface of the laminated 2-in. and 4-in. iron-cork panels, Figs. 20 and 21. No doubt, this is due in part to the fact that these panels had very high heat capacity. It was also more difficult to evaluate accurately the density and specific heat of the cork laminations which were gasket cork and not the variety of cork board used extensively for insulation. Another difficult factor to determine for these panels was the contact resistance between laminations. While the laminations were held firmly together by the weight of the iron for the 4-in. and 8-in. panels, and by bolts outside the center 2-ft square test section for the 2-in. panel, no doubt air films were figured in as cork when calculating density and specific heat. The effect that large air spaces would have on a wall is also a problem with a much more difficult solution.

To give additional information on the error introduced by averaging the physical properties of a compound wall, a few days' data were analyzed from the curve for the top surface temperature in contact with the outside air and the curve for the temperature under the upper heat flow meter. The results of this study are shown in Figs. 22 and 23 for pine and concrete. There was little difference in temperature between the top surface and the surface under the meter for the pine panel, and consequently little difference between the calculated heat flow through the lower surface resulting from these two calculations. The curve calculated from the upper surface temperature is not drawn as it falls practically on the curve calculated from the temperature under the meter. Curves for concrete, resulting from both calculations, are drawn in Fig. 23. In this case, there is considerable difference in temperature between the top surface and the surface under the meter. However, the curves for heat flow as calculated from these two temperature curves check very accurately. The check of the two calculations for gypsum was approximately the same as that for pine and is not shown. Figs. 22 and 23 also include the Fourier analysis curves for the upper surface temperature, curves for the outer air temperature in the shade, and curves of the observed inner surface temperature. It is of interest to note that the temperature of the outside air, observed at a level 12 in. above but in the shade at one side of the top surface of the panel, is considerably below the temperature of the top surface.

EMPIRICAL SOLUTION OF HEAT FLOW USING LIMITED DATA

The rigid mathematical solution of the periodic heat flow problem requires rather laborious and extensive computations to be made on the outside surface temperature curve. It was desirable to decrease the amount of data required for the solution, as well as the labor of making the solution. The similarity of all the top surface temperature curves for the roof panels as shown in Figs. 13 to 23, suggested the possibility of setting up a Fourier series directly from the temperature range between the crest and trough, or the maximum and minimum temperatures, of a given surface for the 24-hr period.

Six outside surface temperature curves for as many different roof panels on two different days are all plotted in Fig. 33. The time of the crest of each curve is plotted as zero time, and the average integrated temperature for the 24-hr period as zero temperature. The superimposed curves give a somewhat consistent series, the characteristics of which are rather accurate functions

of the amplitude or temperature range. It should follow that the constants in the equation resulting from a Fourier analysis of these curves should be approximate functions of the range. The Fourier equation is

$$\theta_{\substack{\text{outer}\\ \text{surface}}} = A \cos a + B \sin a + C \cos 2a + D \sin 2a + + + \qquad (19)$$

where

A, B, C, D are constants to be determined, and a is the timing angle.

Plotting the 6 values of each of the constants, A, B, C, and D, as found by

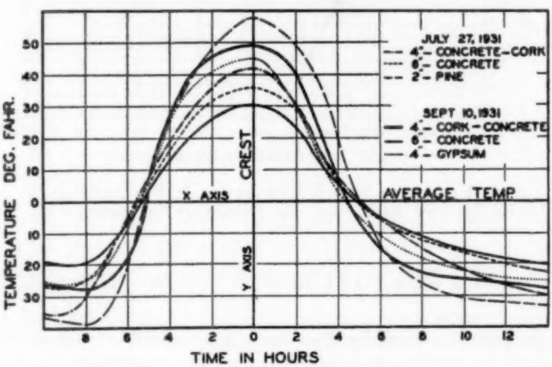


FIG. 33. CURVES FOR UPPER SURFACE TEMPERATURE FOR SEVERAL PANELS SUPERIMPOSED WITH THEIR CRESTS ON THE y AXIS AND THEIR AVERAGE TEMPERATURES ON THE x AXIS

analysis of the 6 curves, Fig. 33, against the range for the respective curves, the points fall fairly close to the curves for these constants in Fig. 34; and hence, to the same degree of accuracy, the constants, for any equation for any day or any roof, within the limits of those included in Fig. 33 may be taken from these curves.

Likewise, the average integrated temperature of the top surface for a 24-hr period may be found by subtracting the temperature difference for the proper range in Fig. 35 from the mean between the maximum and minimum of the outside surface temperatures for the day. In Fig. 24, the use of the empirical method for a determination of an approximation of the Fourier series is illustrated for the 2-in. pine panel on July 28. The crest temperature is 147.3 F, the trough temperature is 59.8 F, the range is 87.5 F, and the average between the crest and trough is 103.6 F. From Fig. 34, the values of the constants A, B, C, and D are therefore: -37.70, $+19.4$, $+3.5$, and -13.8 respectively. From Fig. 35, the difference between the mean of the maximum and minimum temperatures and the average integrated temperature is found to be 7.8 F; and the average integrated temperature is $103.6 - 7.8 = 95.6$ F. The equation for the top surface temperature is

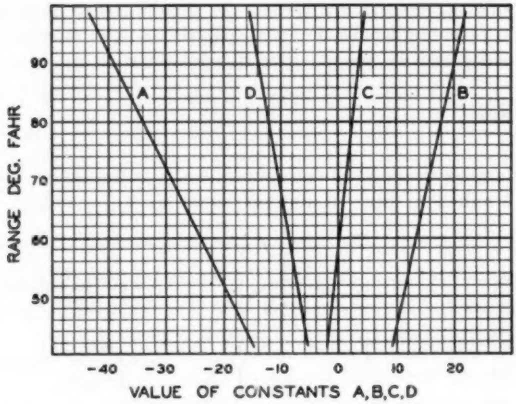


FIG. 34. CURVES GIVING CONSTANTS *A, B, C,* AND *D* IN THE FOURIER EQUATION FOR EMPIRICAL DETERMINATION

$$\theta_{\substack{\text{outer}\\\text{surface}}} = -37.7 \cos a + 19.4 \sin a + 3.5 \cos 2a - 13.8 \sin 2a +++$$

After this temperature equation has been determined, a solution of the heat flow is obtained by continuing from Item 7 of the mathematical solution.

The calculated heat flow through the inner surface, based upon the empirical method, is shown in Figs. 24, 25, and 26 for the pine, concrete, and gypsum panels. These solutions are very satisfactory, and indicate that the method is adaptable for any roof panel within the range of those considered in Fig. 33.

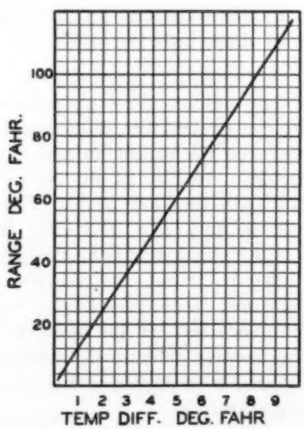


FIG. 35. CURVE GIVING DIFFERENCE BETWEEN THE MEAN OF THE MAXIMUM AND MINIMUM SURFACE TEMPERATURES AND THE AVERAGE INTEGRATED TEMPERATURE FOR EMPIRICAL DETERMINATION

APPLICATIONS TO OTHER THAN HORIZONTAL ROOFS OF SIMPLE HOMOGENEOUS CONSTRUCTION

The data collected and analyzed are for horizontal surfaces only. However, since the primary data considered the outside surface temperature, the methods of calculation presented will apply equally well to structures in any position, as long as the analysis is made of outside surface temperatures.

The curves in Fig. 36 give, *first,* the angles between the direction of the sun's rays and either a horizontal surface, an east, a south, or a west wall; and, *second,* the azimuth angle, or the angle between the direction of the sun's rays and the north. The angles were calculated from data obtained from the American Ephemeris and Nautical Almanac.

The intensity of solar radiation perpendicular to the direction of the sun's rays as observed by the Laboratory pyrheliometer is given in Fig. 37, which represents the maximum intensity of solar radiation observed at the A. S. H. V. E. Laboratory during the three years' study of the subject, and was actually observed on Sept. 7, 1931. In making up the similar curves for July 1 and August 1, in Figs. 38 and 39, the same intensity curve was used by broadening it out for the longer period of sunshine on these days. These curves, therefore, give the maximum solar radiation which may be expected for these months on very clear days with perfect sunshine. Figs. 37 to 39 also show the intensity of solar radiation on a horizontal surface, and on vertical surfaces facing east, south, and west for sun time at Pittsburgh, 40 deg latitude and 80 deg longitude. The intensities on surfaces other than those perpendicular to the sun's rays were calculated from the formula

$$H\phi = H_p \sin \phi \qquad (20)$$

where

$H\phi =$ the intensity of radiation in Btu per hour per square foot on the surface whose angle with the direction of the sun's rays is ϕ.

$H_p =$ the intensity of radiation normal to the direction of the sun's rays, or that observed by the pyrheliometer.

It will be noted that the intensity of radiation on any surface not normal to the sun's rays varies with the month. It should be emphasized that the curves in Figs. 36 to 39, other than those showing the intensity of solar radiation for a surface normal to the sun's rays, will vary with the latitude, and if standard time is used, they will also vary by ± ½ hr, with longitude east and west of the center of the Standard Time Belt. Similar data, collected on the Laboratory recording pyrheliometer, given in Figs. 15 and 16, indicate the variation in solar intensity from day to day.

The circle points in Fig. 37 were taken from the chart of the recording pyrheliometer of the U. S. Weather Bureau as observed on Sept. 7, 1931, at the Pittsburgh Weather Station, approximately five miles from the A. S. H. V. E. Laboratory. The Weather Bureau instrument is unlike either the Smithsonian silver disk instrument or the Laboratory instrument, because it records the intensity of solar radiation on a horizontal surface. Hence, the points from the Weather Bureau record correspond to the curve calculated from the Laboratory pyrheliometer for a horizontal surface, which curve they fit almost

exactly. This is an interesting check for two instruments of widely different design, calibrated entirely independently, and located at stations 5 miles apart in a city.

The main object of the study being to obtain data for typical hot summer

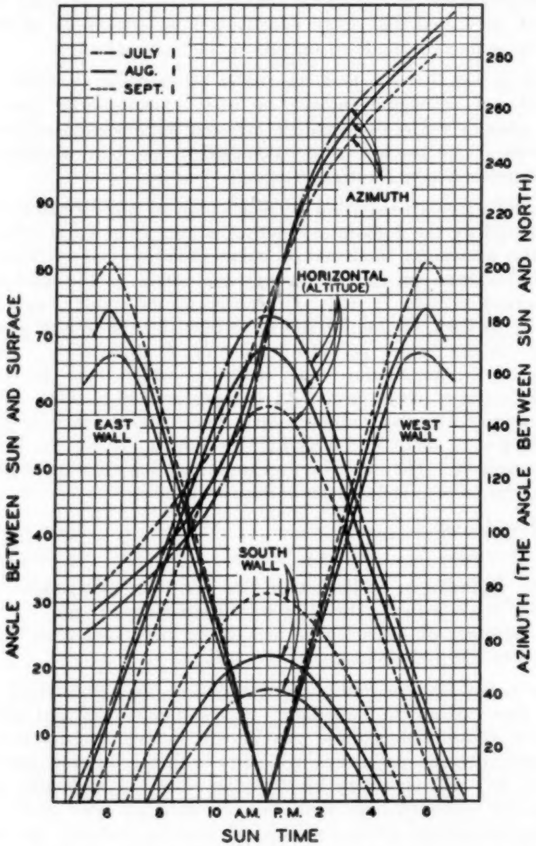


FIG. 36. CURVES GIVING AZIMUTH AND THE ANGLE BETWEEN THE SUN AND A HORIZONTAL SURFACE AND WALLS

days, favorable weather conditions had to be anticipated by careful interpretation of Weather Bureau reports; a very light haze, or a few scattered clouds would make a test erratic. Many data were obtained which had to be discarded because, after several hours of good weather, clouds blanketed the sun, or a thunderstorm suddenly came up and wet the surfaces.

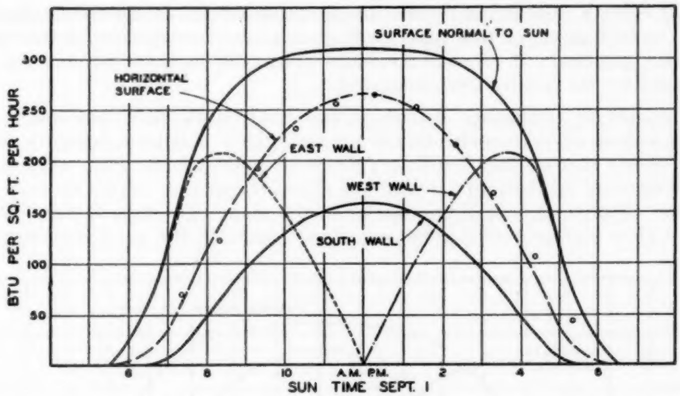


FIG. 37. CURVES GIVING SOLAR INTENSITY NORMAL TO SUN, ON HORIZONTAL SURFACE, AND ON WALLS FOR SEPTEMBER 1. POINTS GIVE WEATHER BUREAU DATA FOR HORIZONTAL SURFACE

An attempt was made to choose test panels which would cover a wide range of typical building construction, and which could be used in the application of the theory. Panels used were either homogeneous or constructed of two homogeneous materials. It was decided not to confuse the work with complex types of materials involving air spaces and non-homogeneous materials which would require a more complicated mathematical solution. The study could well be continued on such complex materials, which would require consideration of confusing factors such as convection currents and additional film resistance coefficients. The determination of proper physical constants for such calculations

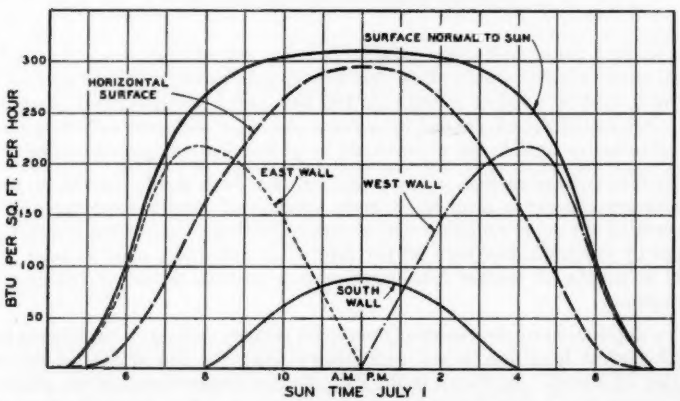


FIG. 38. CURVES GIVING SOLAR INTENSITY NORMAL TO SUN, ON HORIZONTAL SURFACE AND ON WALLS FOR JULY 1

would be very difficult, and many disappointing results should be anticipated. The method given in this paper will give a satisfactory solution for such complex structures if physical constants which will fit the conditions can be assumed for the non-homogeneous panel.

It should be emphasized that the Laboratory's work dealt with roofs and walls without giving consideration to window spaces. Earlier Laboratory tests[8] have shown that on July 1 ordinary single-strength window glass reduces the sun's intensity by about 10 per cent, and allows 90 per cent to pass through and become effective as heat in the interior of the room. The data in Fig. 7 show that a glass surface normal to the sun allows about 275 Btu per square foot per

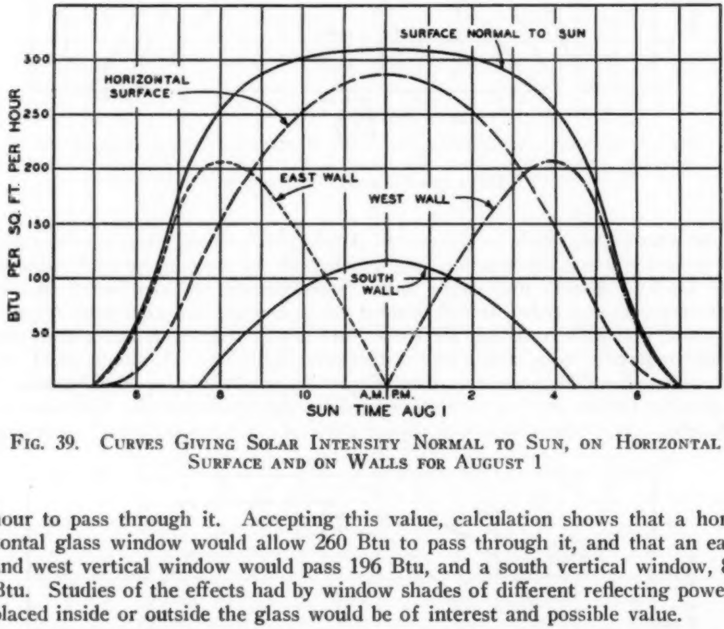


FIG. 39. CURVES GIVING SOLAR INTENSITY NORMAL TO SUN, ON HORIZONTAL SURFACE AND ON WALLS FOR AUGUST 1

hour to pass through it. Accepting this value, calculation shows that a horizontal glass window would allow 260 Btu to pass through it, and that an east and west vertical window would pass 196 Btu, and a south vertical window, 81 Btu. Studies of the effects had by window shades of different reflecting powers placed inside or outside the glass would be of interest and possible value.

The color of the surface is important. As has been pointed out in an earlier Laboratory report,[8] a dead black body which will absorb approximately 100 per cent of the solar radiation will absorb only 70 per cent when covered with a type of red paint, and only 30 per cent when aluminum paint is used. This point is worthy of further consideration as a possible means of reducing heat absorption.

The application of the commonly accepted surface resistance coefficients to the calculation of heat flow is not satisfactory when the sun shines on the outer surface of a wall. As shown in the data, the air temperatures in the shade may be so much lower than the surface temperatures on a hot sunny day because of

[8] See Bibliography, *d*.

solar radiation that the heat flow would actually be positive, although the surface coefficients as ordinarily used might indicate a negative heat flow.

BIBLIOGRAPHY

a Coefficients of Heat Transfer as Measured Under Natural Weather Conditions, by F. C. Houghten and C. G. F. Zobel (A. S. H. V. E. TRANSACTIONS, Vol. 34, 1928).

a Heat Transfer Through a Roof Under Summer Conditions, by F. C. Houghten and C. G. F. Zobel (A. S. H. V. E. TRANSACTIONS, Vol. 34, 1928).

a Additional Coefficients of Heat Transfer as Measured Under Natural Weather Conditions, by F. C. Houghten, Carl Gutberlet and C. G. F. Zobel (A. S. H. V. E. TRANSACTIONS, Vol. 35, 1929).

b Measuring Heat Transmission in Building Structures and a Heat Transmission Meter, by P. Nicholls (A. S. H. V. E. TRANSACTIONS, Vol. 30, 1924).

c Mathematical Theory of Heat Conduction, by L. R. Ingersoll and O. J. Zobel.

d Absorption of Solar Radiation in Its Relation to the Temperature, Color, Angle and Other Characteristics of the Absorbing Surface, by F. C. Houghten and Carl Gutberlet (A. S. H. V. E. TRANSACTIONS, Vol. 36, 1930).

DISCUSSION

J. A. GOFF[9] (WRITTEN): On reading this elaborate report it is felt that the authors have not presented the results of their investigation in the simplest and most understandable form, although the experimental and analytical information which the report contains is extremely interesting and valuable. Instead of directing especial attention to the elements of the problem by applying their mathematical analysis to the simplest examples. the authors have passed almost at once to the most complicated examples and discuss in detail methods of calculation which are really incidental to the main purpose of the report. Also, the inclusion of too many graphs and tables serves to complicate rather than to simplify the presentation of results.

The mathematical analysis on which the authors have based their study of the periodic flow of heat through a wall or slab only applies strictly to what may be called an ideal wall or slab satisfying certain definite specifications as follows:

(1) The material of which the slab is composed is homogeneous and isotropic as regards the flow of heat and its thermal properties can be described by certain characteristic quantities (conductivity k, density ρ, and specific heat c) which are susceptible of experimental measurement.

(2) The slab is bounded by two parallel planes of infinite extent a finite distance l apart which is the thickness of the slab. Under these conditions the isothermals are known in advance to be planes parallel to the bounding surfaces so that the flow of heat through the slab can be treated as one-dimensional.

(3) The lower surface of the slab dissipates heat at a rate proportional to the excess of its temperature above that of its surroundings (other conditions affecting the dissipation of heat being constant), the proportionality factor being a property of the surface called its emissivity ϵ which is also susceptible of experimental measurement.

(4) The temperature of the upper surface varies periodically with the time due presumably to its exposure to solar radiation. In the simplest case this variation would be purely sinusoidal with a period of $\frac{2\pi}{\omega}$ hours.

The differential equation (1) of the authors' paper states that the rate at which heat flows across any parallel surface within the ideal slab is proportional to the temperature gradient at that surface, the proportionality factor being the conductivity k of the slab material (assumed constant). Also, the accumulation of heat

[9] Associate Professor of Thermodynamics, University of Illinois.

at any point within the slab produces an increase of temperature at that point which is determined by the specific heat c and the density ρ of the slab material.

This differential equation together with the appropriate boundary conditions stated in paragraphs 3 and 4 of this discussion yields a unique solution of the problem which is expressed accurately by the authors' equation (5). This equation gives the temperature variation (with time) at any distance x below the upper surface and, for $x = l$, at the lower surface itself.

The simplest problem in the periodic flow of heat through an ideal slab of thickness l together with its solution may now be stated in more practical terms as follows. Suppose that the lower surface of the slab is surrounded by air at a constant temperature t_0 (other conditions which affect the emissivity ϵ remaining constant also) and that by exposure to solar radiation, its upper surface temperature experiences a *sinusoidal* variation rising θ_0 degrees above and falling θ_0 degrees below an average value of t_1 degrees. Then the temperature at the lower surface should rise $M\theta_0$ degrees above and fall $M\theta_0$ degrees below an average temperature of $t_0 + N(t_1 - t_0)$ degrees. Moreover, the maximum lower surface temperature should occur γ hours after the maximum upper surface temperature.

The quantities M, N and γ are fully determined from the following equations in which the authors' notation is used so far as possible.

$$M = \frac{e^{-\beta l}}{B}\sqrt{1 + A^2 + 2A\cos\sigma} \tag{1}$$

$$N = \frac{1}{1 + \frac{\epsilon}{\kappa} l} \tag{2}$$

$$\gamma = \beta l - \alpha - \zeta \tag{3}$$

the angle ζ being given by

$$\sin\zeta = \frac{A\sin\sigma}{\sqrt{1 + A^2 + 2A\cos\sigma}} \qquad 0 \le \zeta \le \frac{\pi}{2} \tag{4}$$

Thus, for any slab material whose thermal properties (conductivity k, specific heat c, density ρ and emissivity ϵ) are known, curves showing how these quantities M, N and γ vary with the thickness l of the slab can be drawn. The curves so drawn must necessarily refer to a particular period $\frac{2\pi}{\omega}$ which in the simplest case would be 24 hours for the fundamental of the actual upper surface temperature variation.

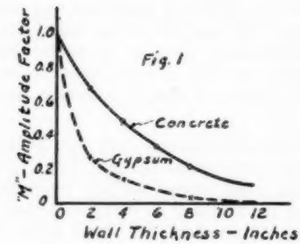
Figs. 1, 2, and 3 relate to the gypsum and concrete panels whose thermal properties have been listed by the authors. They apply only for the fundamental of the actual temperature wave with an assumed period of $\frac{2\pi}{\omega} = 24$ hours.

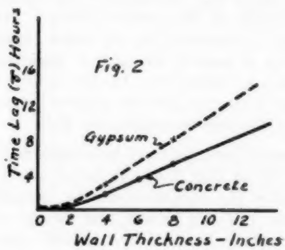
For the purpose of illustrating the use that can be made of these curves, the data represented in Fig. 26 of the paper may be analyzed. Fig. 26 relates to the 4-in. Gypsum Panel (actual thickness 4.2 in.) and shows a *maximum* temperature at the upper surface of 134 deg and an *average* temperature of 87 deg. The air temperature below the panel is $t_0 = 69.6$ deg. The correct values of M, N and γ read from Figs. 1, 3 and 2 respectively are $M = 0.150$, $N = 0.155$ and $\gamma = 3.2$; hence, the lower surface temperature will vary $0.150(134 - 87) = 7.1$ deg above and below an average of $69.6 + 0.155(87 - 69.6) = 72.3$ deg. Moreover, it reaches a maximum value of $72.3 + 7.1 = 79.4$ deg just 3.2 hours after the maximum upper surface temperature. The maximum heat flow out of the lower surface would be $1.9(79.4 - 69.6) = 18.6$ Btu per sq ft per hour.

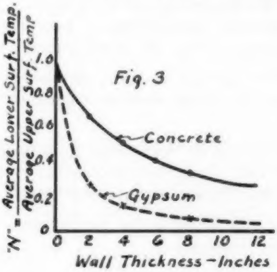
Comparing the above calculations with the measured results (as shown in Fig. 26) would seem to indicate that so far as locating the time of occurrence of the maximum lower surface temperature is concerned, it is hardly necessary to analyze

the actual upper surface into fundamental and first harmonic according to the methods of the Fourier analysis. On the other hand the determination of the maximum heat flow is appreciably higher than the measured 16 Btu per sq ft per hour.

In connection with Fig. 2 it is interesting to note that for slabs of thickness greater than about 3 in. the time lag between maximum upper surface temperature







(Calculated Points)

l Inches	"M" Amplitude Factor		(γ) Hours		"N" = Ave. Lower Surf. T. / Ave. Upper Surf. T.	
	Concrete	Gypsum	Concrete	Gypsum	Concrete	Gypsum
2	0.6737	0.2738	0.568	0.715	0.677	0.276
4	0.4801	0.1408	1.735	2.930	0.513	0.160
6	0.3306	—	3.530	—	0.412	—
8	0.2180	0.0376	5.250	8.100	0.345	0.087
16	0.0403	0.0025	11.980	18.500	0.208	0.0323

CURVES FOR CONCRETE AND GYPSUM PANELS

and maximum lower surface temperature is directly proportional to the thickness. For the linear portions of these curves the following equations apply:

$$\text{Concrete:} \quad \gamma = 0.83\,l - 1.3$$
$$\text{Gypsum:} \quad \gamma = 1.29\,l - 2.2$$

γ being given in hours and l in inches.

Of particular interest is Fig. 19 which relates to the 4 in. concrete-cork panel and shows, as pointed out by the authors, poor correlation between observed and calculated results. This discrepancy indicates that the simple analysis of an homogeneous slab does not apply to a composite slab such as the one under consideration. To apply the simple analysis of an homogeneous slab, one would naturally determine certain average quantities for the density, conductivity and specific heat of the composite slab. Thus, imagine a cylindrical sample taken perpendicular to and including the surfaces of the slab and of such cross-section that its volume is exactly one cubic foot; then the weight of the sample may be called the *density,* the ratio of the heat capacity of the sample to its weight, the *specific heat,* and the conductance of the sample multiplied by its length, the *conductivity* of the composite slab.

Numerical results were computed for the particular concrete-cork slab B′ shown in Fig. 2 of the paper. A pure sinusoidal temperature variation at the upper surface of 53 deg above and 53 deg below an average of 27 deg (the air temperature below

the slab being called 0 deg, was assumed. By using average quantities determined in the manner described in the preceding paragraph and substituting them into the simple formulae for the homogeneous slab, the following results were obtained. The lower surface temperature should vary 2.23 deg above and below an average of 3.73 deg, its maximum value occurring 9.8 hours after the maximum upper surface temperature.

Now a correct analysis of the composite slab would have to take account of the discontinuity at the contact surface between the two different materials and would naturally be expected to be more complicated than the analysis of the homogeneous slab. As a matter of fact it involves the simultaneous solution of twelve linear equations to determine twelve arbitrary constants so that it is a question as to whether it is possible to express the results by means of simple formulae similar to those for the homogeneous slab.

Numerical results have been obtained for the particular concrete-cork slab described above (B′ in Fig. 2 of the paper) by H. B. Goff, of Colorado Springs, Colo., working under the direction of the writer. For the same upper surface temperature variation (53 deg above and below an average of 27 deg) it turns out that the lower surface temperature should vary 5.83 deg (instead of 2.23 deg) above and below an average of 3.73 deg (same as before), its maximum value occurring 4.23 hours (instead of 9.8) after that of the upper surface temperature. These results are in good agreement with the measured results shown in Fig. 19 of the paper.

In conclusion, the writer wishes to congratulate the authors on the valuable experimental and analytical information which they have presented in their report on the periodic flow of heat.

L. A. HARDING (WRITTEN): The engineer is primarily interested in a method for approximating the maximum possible rate of heat flow into a building due to solar radiation for a given latitude in advance of the design of the structure.

The writer is of the conviction that this is entirely feasible, as he believes will be apparent from the following brief resumé of the problem:

(1) It is obvious that the maximum rate of heat flow into a building will occur when the solar heat cycle is repeated for a number of successive clear sky days.
(2) With a repeated solar heat cycle and for any particular type of construction with a constant air temperature condition maintained inside the building, the cycle of surface temperature and heat flow are also repeated when the outside wind movement is normal and therefore, definite ratios exist between the maximum rate of solar radiation, maximum outside surface temperature and the maximum rate of heat flow for a specific construction.
(3) The physical characteristics of the surfaces and the materials employed remain constant to all intents and purposes, therefore it may be safely assumed that the above mentioned ratios are a function of certain combinations of these characteristics.

The writer proposes the following formulae for determining the *maximum* rate of heat flow through a horizontal roof based on a maximum observed outside roof surface temperature and the physical characteristics of the materials composing the roof deck construction.

f = inside surface coefficient Btu per square foot per degree difference.
t_1 = maximum inside surface temperature degrees Fahrenheit.
t_a = temperature air inside degrees Fahrenheit.
h = maximum rate of heat flow into building Btu square foot per hour.
$= f(t_1 - t_a)$ .... [1]
k = coefficient of conductivity.
c = specific heat roof deck.
p = density roof pounds cubic foot.

t_0 = maximum outside surface temperature degrees Fahrenheit.
R = a constant—depending primarily on the coefficient of absorption of the roof surface.

$$h = f\left(Rt_0 \sqrt[4]{\frac{k}{cp}} - t_a\right) \quad [2]$$

The values of R as determined by the writer from the test data submitted by the authors of the paper follow.

DATA FROM TESTS

Material	$\sqrt[4]{\frac{k}{cp}}$	t_0	h	t_a	f*	t_i	R
6.188 in. Concrete	0.69	119.0	32.2	70	1.9	86.9	1.06
2.156 in. Pine	0.47	149.7	25.1	70	1.9	83.0	1.18
4.188 in. Gypsum	0.55	132.0	18.5	70	1.9	79.7	1.10
4.188 in. Concrete-Cork	0.49	139.4	16.5	70	1.9	78.7	1.16
4.188 in. Cork-Concrete	0.49	134.6	10.24	70	1.9	75.4	1.17
					Average of above tests =		1.13
2.67 in. Iron-Cork	0.44	151.4	9.7	70	1.0	79.7	1.20
4.47 in. Iron-Cork	0.39	148.7	6.17	70	1.25	74.9	1.29
8.32 in. Iron-Cork	0.41	145.2	6.00	70	1.25	74.8	1.25

* Lamp black coated surface.

In practice t_0 for horizontal roofs has been observed to reach a maximum of 160 to 170 F in northern states.

The value of R for dark surfaced roofing material such as asphalt surfaced roofing, slate or black iron will probably not exceed 1.2. The value of f may be assumed as 1.4; t_0 may be assumed as 160 deg. Substituting these values in the above equation

$$h = 268 \sqrt[4]{\frac{k}{cp}} - 1.4\, t_a \quad [3]$$

It is believed the employment of this formula will produce safe and satisfactory results for the purposes intended.

The heat flow due to solar radiation through horizontal and vertical glass surfaces and the effect of white shades has been previously discussed by laboratory papers.

The paper presented at this meeting by Messrs. Walker, Sanford and Wells provides sufficient data for the engineer to estimate with reasonable accuracy the heat flow due to solar radiation through vertical glass surfaces.

R. E. BACKSTROM (WRITTEN): The authors have been most thorough in the treatment of a complicated problem and as a result have developed data which will serve as a basis for further research. The report shows how heat flow at any instant can be calculated from certain known facts, including the outside surface temperature. This is a long step toward the ultimate solution of the problem. However, the determination of the outside surface temperature from the known intensity of solar radiation, air temperature, and wind velocity is a problem which must be solved before the data can be easily applied. It is my suggestion that this paper be considered a progress report and the work be continued with a view of obtaining a more practicable solution.

A. R. STEVENSON: Two points were of particular interest to me. First, the fact that flow of heat is identical with flow of electricity and can be analyzed by iden-

tical mathematical processes, heat capacity being equivalent to electrical capacity and thermal resistance analogous to electrical resistance. Second, the fact that the time lag between the temperature impulse on one side of a wall is related to the resulting heat flow on the other. It was observed several years ago in testing refrigerator boxes that a temperature change in the room in which the boxes were being tested resulted in a similar change inside the box about four hours later.

Likewise, maximum outside temperature at noontime may result in peak flow of heat to the inside of a room in the evening. This peak cooling load may be provided for by starting the cooling system in the early afternoon and thereby removing heat from the walls in order to help take care of the maximum evening cooling load.

PRESIDENT CARRIER: I would like to hear discussion pertaining to the winter heating load as affected by temperature changes.

E. K. CAMPBELL: The application of the results of this paper to the intermittent heating or cooling of a building, such as a church, is of value. By blowing cool air through a church during the night and keeping the building closed after sunrise, the temperature of the inner contents of the building will be lowered considerably and they will absorb heat given off by the audiences during services until about noon. This will maintain a comfortable condition approximating that obtained by refrigeration.

PRESIDENT CARRIER: I have visualized the application of these results as very important in winter heating. We have exact coefficients of conductivity and heat transfer, but their applications do not really mean a thing excepting for thin materials having small heat capacity such as glass. With thicker walls having greater heat capacity, the varying temperature prevailing results in a maximum rate of heat penetration which is considerably lower than that calculated from data available in the past. It is necessary to use a lower rate of heat flow to compensate for the heat capacity. The application of our transmission tables or perhaps the tables themselves will have to be revised in order to give accurate rates of heat flow for every construction.

F. C. HOUGHTEN: The authors are interested in the further analysis of the problem presented by Professor Goff. It should be pointed out, however, that the purpose of the paper was not only to attempt a mathematical solution, but also to present data in the form of curves which could be applied directly in practice by analogy. This is the reason for the inclusion of the large number of curves for the various types of construction tested.

The authors agree with the opinion frequently expressed that a shorter solution, even though not rigidly accurate, is desirable, and it is possible that Mr. Harding's presentation may be used as such. However, a rigid mathematical solution must be used in order to determine the accuracy of the application of any short cut.

No. 924

FIELD STUDIES OF OFFICE BUILDING COOLING

By J. H. Walker,[1] S. S. Sanford[2] (*MEMBERS*) and E. P. Wells[3] (*NON-MEMBER*) Detroit, Mich.

Report of research conducted in co-operation with the Committee on Research of the American Society of Heating and Ventilating Engineers

THERE is a tendency in ventilating system design to copy previous work without an adequate check-up of actual performance. A great many mistakes could doubtless be avoided, particularly in a relatively new field such as building cooling, if design methods were carefully correlated with field experience.

The tests described in this paper were made because the authors felt that some actual results of the performance of a building cooling system would be of considerable value to designers of such systems. Because the tests were conducted under conditions which approximated ordinary operating conditions, it was impossible to control some of the factors as accurately as would be done in a laboratory test, and this should be borne in mind in making use of the results.

The work was done in the Union Guardian Building in Detroit, shown in Fig. 1. It is a 40-story building of which the lower 16 floors and two basements are artificially cooled. The air-conditioning system has been previously discussed,[4] but the following is a brief description of the important features. The building is of more or less typical office building construction having brick walls, backed up with hollow tile, and with two large steel-sash windows per bay. The heating as well as the cooling requirements are taken care of by a fan system. Air enters each room through one or more ceiling diffusers which

[1] Superintendent of Central Heating, The Detroit Edison Company.
[2] Sales Engineer, The Detroit Edison Company.
[3] Air Conditioning Engineer, Union Guardian Building.
[4] Air Conditioning System of a Detroit Office Building, by H. L. Walton and L. L. Smith (A. S. H. V. E. Transactions, Vol. 35, 1929); also, Operation of an Air Conditioning Plant in a Large Office Building, by Earl P. Wells (*Heating & Ventilating*, September, 1931, p. 62).

Presented at the 38th Annual Meeting of the American Society of Heating and Ventilating Engineers, Cleveland, Ohio, January, 1932, by S. S. Sanford.

FIG. 1. UNION GUARDIAN BUILDING

discharge horizontally in all directions except towards the windows and the exhaust is taken care of by grilles in the office partitions through which the air finds its way to the corridor and thence to the two recirculating risers. The system is zoned and each zone is equipped with a variable speed fan, making it possible to control the temperature on different sides of the building in summer

by varying the volume of air supplied. Data regarding the building and air conditioning system are given in Table 1.

TABLE 1. UNION GUARDIAN BUILDING—PHYSICAL DATA

Building height	40 stories
Ground area	80 ft x 270 ft
Air-conditioned floors	16 lower floors and 2 basements
Air-conditioned volume	2,962,600 cu ft including toilets, inside stairs, and elevator shafts
Air-conditioned area	225,986 sq ft rentable area
Number of occupants in air-conditioned area	1850
Refrigeration capacity	600 tons
Compressors	3—200 ton CO_2
Compressor motors	3—300 hp 4600 v direct drive, synchronous
Dehumidifiers	5 with direct expansion evaporating coils
Pumps	5 spray and 5 flood
Fans	10 circulating (no general exhaust fans)
Condensers	Double-pipe, using city water

The maximum weather conditions in Detroit for which cooling systems are designed are approximately 95 F dry bulb and 75 F wet bulb. The Union Guardian Building is used for banking, trust company and tenant office purposes. The temperature carried on the office floors varies from 73 F dry bulb and 60 F wet bulb to 79 F dry bulb and 64 F wet bulb, depending upon the outside temperature. The temperature in the lobby is about one degree higher than in the offices and that in the basements about two degrees lower than in the offices.

The various factors which comprise the cooling and dehumidifying load of a building are as follows:

I. Load due to weather conditions:
   1. Heat transmission through walls and roof.
   2. Heat transmission through window glass.
   3. Sun radiation through window glass.
   4. Heat and moisture in make-up and leakage air.

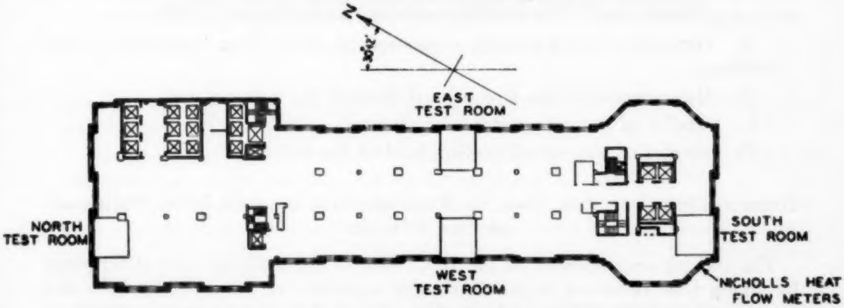


FIG. 2. FLOOR PLAN OF BUILDING SHOWING TEST ROOMS

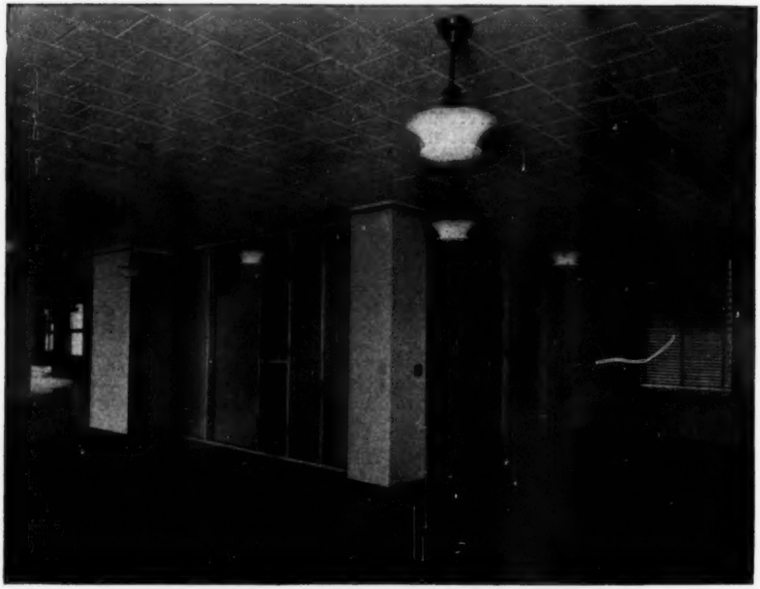


FIG. 3. EXTERIOR OF TEST ROOM

II. Load due to occupancy:
   5. Heat from lights.
   6. Body heat and moisture from occupants.
   7. Heat and moisture from machinery, piping and processes.

The tests herein described were confined to one factor in the total cooling load of the building, namely, that due to external weather conditions, and involve items 1, 2 and 3, which are the items of perhaps the greatest uncertainty in cooling calculations. The studies comprised several phases as follows:

A. Determination of cooling requirements of the four exposures of the building.

B. Measurement of the flow of heat through the building wall.

C. Studies of the temperatures of the wall.

D. Studies of the overall cooling load of the building.

DETERMINATION OF THE COOLING REQUIREMENTS OF THE FOUR EXPOSURES OF THE BUILDING

The cooling requirements of the four sides of the building were determined by taking four rooms of identical size and exposure on the different sides and measuring the temperature and quantity of cooled air which was required to maintain the rooms at an equal and constant temperature throughout the work-

ing day. These rooms were specially constructed test rooms on the 12th floor of the building and were located as shown in Fig. 2.

They were approximately 14 ft 8 in. wide and 17 ft deep, taking in almost the width of a bay of two windows, and had walls made of ½ in. rigid insulation nailed to wooden studding. The exterior of one of the rooms is shown in Fig. 3. There were two openings in each room, one being a small doorway through which observers entered the room and which was kept closed except when someone was passing through it. The other opening, located about 18 in.

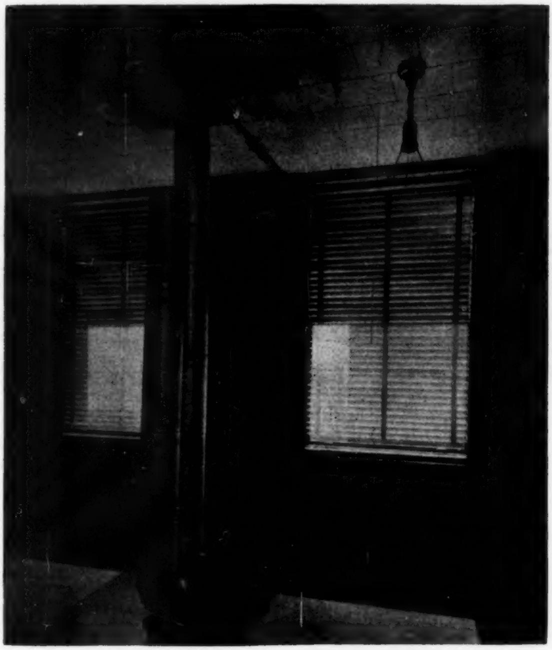


FIG. 4. INTERIOR OF TEST ROOM

from the ceiling and to one side of the door, was the air outlet from the room and was approximately 21 in. x 12 in.

The interior of one of the test rooms is shown in Fig. 4, and a section and the elevation of the outside wall in a test room is shown in Fig. 5. The windows are equipped with Venetian blinds which were adjusted during the tests so as to cover the upper half of the window, with the slats in a horizontal position. The windows are of plate glass ¼ in. thick, and the window frames and sash are of steel.

Normally, air is delivered to the offices through square ceiling diffusers, two to each bay. In the test rooms, the ceiling diffusers were removed because it

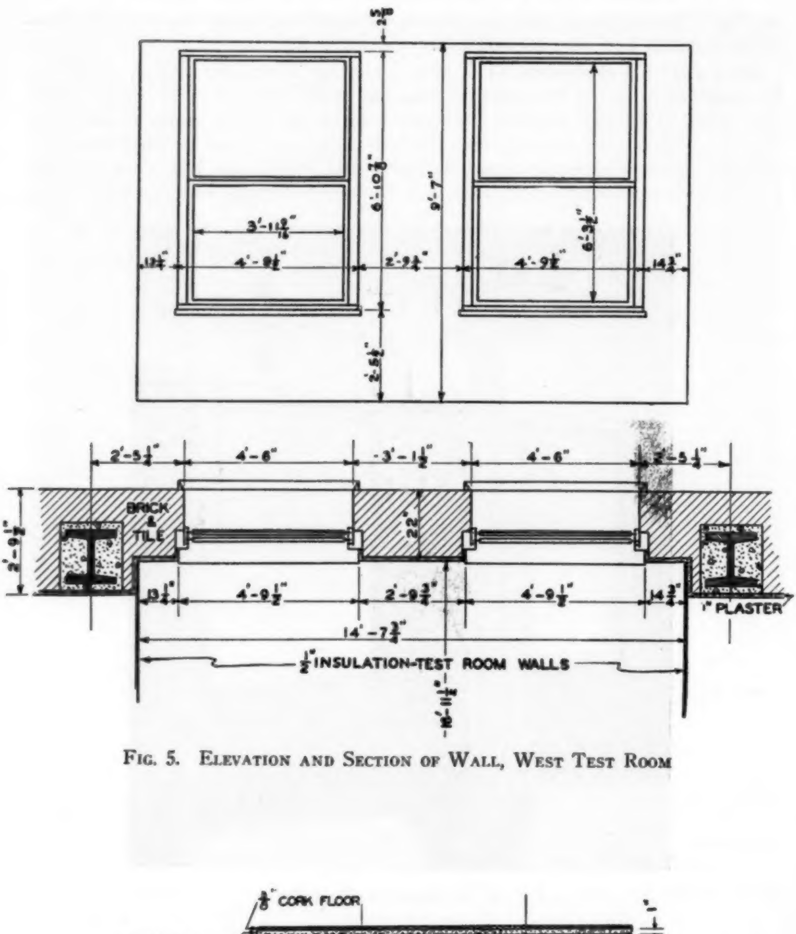


FIG. 5. ELEVATION AND SECTION OF WALL, WEST TEST ROOM

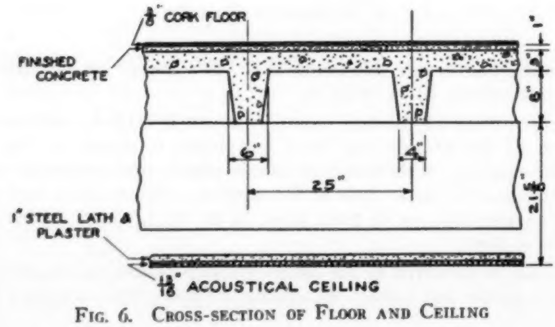
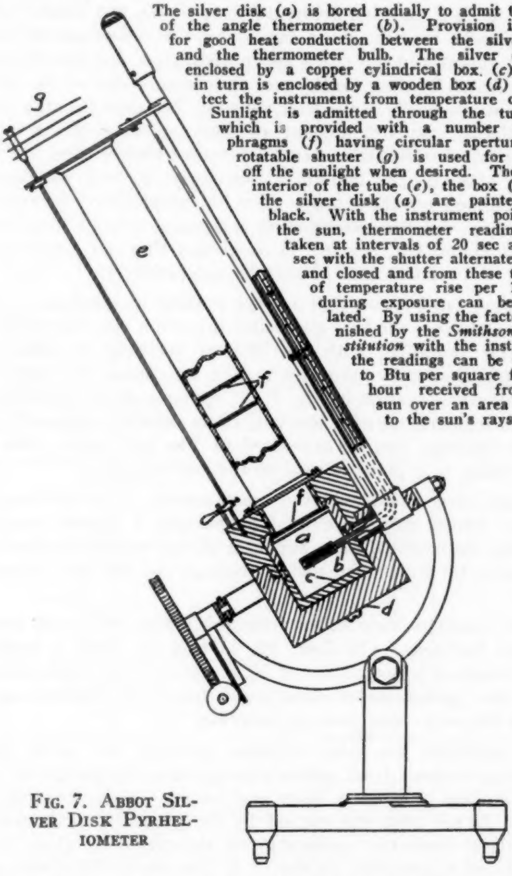


FIG. 6. CROSS-SECTION OF FLOOR AND CEILING

was difficult to regulate and measure the air coming from them and one of the openings in the ceiling was plugged. An 8 in. circular sheet-iron duct with a butterfly damper in it was attached to the other ceiling opening and all of the air was delivered to the room through this. The duct extended to the floor



The silver disk (*a*) is bored radially to admit the bulb of the angle thermometer (*b*). Provision is made for good heat conduction between the silver disk and the thermometer bulb. The silver disk is enclosed by a copper cylindrical box (*c*) which in turn is enclosed by a wooden box (*d*) to protect the instrument from temperature changes. Sunlight is admitted through the tube (*e*) which is provided with a number of diaphragms (*f*) having circular apertures. A rotatable shutter (*g*) is used for cutting off the sunlight when desired. The entire interior of the tube (*e*), the box (*c*), and the silver disk (*a*) are painted dead black. With the instrument pointed at the sun, thermometer readings are taken at intervals of 20 sec and 100 sec with the shutter alternately open and closed and from these the rate of temperature rise per 100 sec during exposure can be calculated. By using the factors furnished by the *Smithsonian Institution* with the instrument, the readings can be reduced to Btu per square foot per hour received from the sun over an area normal to the sun's rays.

FIG. 7. ABBOT SILVER DISK PYRHELIOMETER

and back toward the ceiling as shown in Fig. 4, and the volume of air was determined by an anemometer held in the outlet of the duct.

The floor and ceiling construction in the building is as shown in Fig. 6. In the space between beams are the air ducts. The temperature in the furred space above the test rooms was read with thermometers thrust up through holes drilled in the ceiling; that in the furred space beneath the test rooms was obtained by thermometers through the ceiling below. The temperature outside the building wall was obtained from a thermocouple shielded by an insulated

container having a bright metal surface with the container arranged for a free flow of air up through it.

The temperature of the entire twelfth floor was maintained as nearly as possible equal to that in the test rooms. The temperature above the ceiling ranged from 72 to 76 F and that below the floor of the test rooms ranged from 74 to 76 F. Except for the error caused by the difference in temperature between the test rooms and the space above and below, the heat removed from the room, as indicated by the quantity and temperature rise of the air supplied, should be substantially equal to the heat entering the room through the exposed side. In calculating the net amount of heat entering the test rooms, a correction was made for differences in temperature between the test rooms and the surrounding space in the building. Obviously, no extreme degree of accuracy can be claimed for these tests. The air temperature through the room varied somewhat as did the surrounding temperatures and there were other small sources of error; but, bearing in mind that they are field tests and not laboratory tests, the method used should be acceptable.

It was, of course, anticipated that a large portion of the load would be due to sunshine through the window glass and provision was therefore made to measure the solar radiation, with the idea of arriving at some correlation between solar radiation and cooling load. The instrument used was the Abbot pyrheliometer, shown in section in Fig. 7. It operates on the principle of measuring the rise in temperature of a block of silver which is exposed to the sun's radiation. The readings can be expressed in Btu per square foot per hour, the area being taken in a plane normal to the sun's rays.

After the tests were completed the anemometers were calibrated for the conditions under which they were used by passing a known quantity of air through a circular duct which was a duplicate of that used in the test rooms and noting the relation between anemometer readings and the true quantity of air flowing.

Pyrheliometer readings were taken from the top of the south tower of the Union Guardian Building. On Sept. 10, 11 and 12, 1931, a series of days with maximum outdoor temperatures of 90 F and over, determinations of solar radiation with the pyrheliometer were made frequently throughout each day during the time the room tests were in progress.

In order to determine the solar radiation entering the rooms through the window glass, observations from within a room were made with the pyrheliometer with the window alternately open and closed. These readings indicated that from 12 to 20 per cent was cut off by the glass. Experiments by F. C. Houghten and Carl Gutberlet [5] indicated an absorption by glass of from 8.9 to 16.5 per cent and experiments by the U. S. Bureau of Standards referred to in the same paper showed the absorption of window glass to be 18 per cent. Of the radiation which does not pass directly through the glass, part is absorbed and part reflected. Of that which is absorbed, part is delivered by the glass to the room. It was therefore assumed that of the total solar radiation, 93 per cent was delivered to the room and the other 7 per cent reflected or transferred by the glass to the outdoor air.

[5] Absorption of Solar Radiation in Its Relation to the Temperature, Color, Angle and Other Characteristics of the Absorbing Surface, by F. C. Houghten and Carl Gutberlet (A. S. H. V. E. TRANSACTIONS, Vol. 36, 1930).

Results of Tests

Pyrheliometer readings for Friday, Sept. 11, 1931, are shown in Fig. 8. The time shown in this figure and throughout the paper is Eastern Standard (75th Meridian) time.

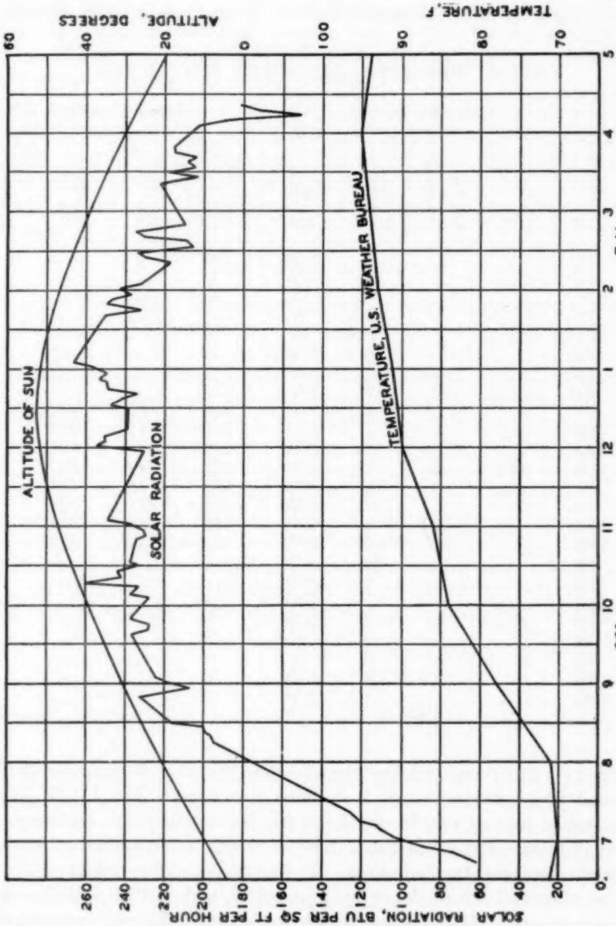


Fig. 8. Solar Radiation and Outdoor Temperature on a Clear Day, Sept. 11, 1931

The results of the cooling tests in the 4 test rooms for Sept. 11, 1931, are given in Tables 2 and 3 and in Fig. 9. Similar tests were made on other days but the data for Sept. 11 are especially important because this was a day of continuous bright sunshine. In Fig. 9 the data are plotted separately for each

test room. The *actual cooling load* is the rate of removal of heat from the room by the cooling air as determined by the anemometer readings and the rise in temperature of the air in passing through the room. As defined here it did not include the dehumidifying load due to moisture from people and from outdoor air. The volume and temperature of the air are given in Table 3. *Conduction, glass* is the calculated rate of heat transmission through the

Table 2. Atmospheric Conditions, Sept. 11, 1931

Eastern Standard Time	U. S. Weather Bureau					Solar Radiation (Btu per Sq Ft per Hour)	Indoor Conditions, 12th Floor		
	Temperature (F)	Wet Bulb (F)	Relative Humidity (Per Cent)	Wind Velocity (mph)	Wind Direction		Dry Bulb (F)	Wet Bulb (F)	Relative Humidity (Per Cent)
1 a.m.	78	....	..	9	S W	0	....	....	..
2	78	....	..	10	S W	0	....	....	..
3	78	....	..	9	W	0	....	....	..
4	76	....	..	9	W	0	....	....	..
5	74	....	..	8	W	0	....	....	..
6	72	....	..	8	S W	0	....	....	..
7	70	....	..	9	S W	95	....	....	..
8	71	67.4	83	11	S W	173	....	....	..
9	78	....	..	8	S W	220	74.5	62.5	51
10	84	....	..	7	S W	235	74.5	63.0	53
11	86	....	..	7	S W	235	74.0	62.0	51
12	90	74.1	48	9	S W	245	74.5	61.5	48
1 p.m.	91	....	..	11	S W	255	75.0	62.0	48
2	93	....	..	12	S W	240	75.0	62.0	48
3	94	....	..	9	S W	215	75.0	62.0	48
4	95	....	..	11	S W	197	74.0	62.0	50
5	94	....	..	10	S W	...	....	....	..
6	93	....	..	9	S W	...	....	....	..
7	90	....	..	7	S W	0	....	....	..
8	87	73.0	50	9	S W	0	....	....	..
9	84	....	..	9	S	0	....	....	..
10	80	....	..	8	S	0	....	....	..
11	80	....	..	8	S W	0	....	....	..
12	80	....	..	6	S W	0	....	....	..

glass in the two test room windows based upon Weather Bureau temperatures and observed room temperatures 5 ft from the floor. The U. S. Weather Bureau station is located two blocks from the Union Guardian Building. *Sunshine through glass* is the calculated rate at which solar radiation enters the test rooms through the two windows. It was obtained by reducing the solar radiation as observed out-of-doors by 7 per cent and multiplying this by the area of sunshine on a plane normal to the sun's rays. The *net calculated load* is the algebraic sum of the heat entering the room by conduction through the glass, solar radiation through the glass, transmission through the outer wall, heat from the observers during the few minutes they were in the test room and heat entering or leaving the test room because of difference in temperature between the room and the surrounding space in the building.

It is noteworthy that the amount of heat entering through the wall was too small to plot in Fig. 9. The Nicholls' heat meter readings for this day on a southerly-exposed wall indicated that the flow of heat from the wall to the room was only about 1.7 Btu per square foot per hour when the room temperature was 75 F. On this basis the heat from the wall to the room was calculated to be 125 Btu per hour in the east and west rooms and 135 Btu per hour in the north and south rooms, these rooms having a slightly larger wall area. The heat from the wall was assumed constant although it did appear to decrease somewhat during the day.

No allowance was made for infiltration. On the day of the test the wind velocity did not exceed 12 mph and it was assumed that the pressure in the test rooms was sufficient to prevent infiltration. Air which may have leaked outward was assumed to have the same temperature as that leaving the room through the outlet and was included on that basis in the computation of the cooling load.

DISCUSSION OF RESULTS

The rapid fluctuation in intensity of solar radiation shown in Fig. 8 may be attributed partly to changes in atmospheric conditions and partly to errors in observation. The seasonal variations in solar radiation do not appear to be great if days having a clear bright sky are compared. Readings taken during the middle of the day over the period from Aug. 15 to Nov. 13 show little change. Seasonal variations in the position of the sun do, however, have an effect on the solar radiation entering through the windows. On June 21 with the sun high in the sky the normal area of sunlight entering through a window is a minimum. Likewise the normal area is a maximum on Dec. 21. It follows that flat roof surfaces of a building receive the maximum amount of radiation in June whereas walls and windows receive their maximum later in the year. At noon in the latitude of Detroit the amount of heat received by vertical and horizontal surfaces on June 22 and Sept. 11 expressed as a percentage of that received by a surface normal to the sun's rays is as follows:

	Vertical Surface Per Cent	*Horizontal Surface Per Cent*
June 22	30.6	95.1
Sept. 11	60.8	79.3

Fig. 9 shows that solar radiation through the window glass accounts for the largest part of the cooling required in the test rooms. The north room which received no sun needed a very small amount of cooling compared with that required in the other rooms. The effect of sunshine is especially evident in the west room where the amount of cooling required increased rapidly shortly after the sun began to shine in through the windows. In the latter part of the afternoon when the building across the street shaded this room, it was necessary to reduce immediately the amount of cooling air to prevent the room temperature from falling.

The large amount of heat entering a building through the windows in the form of solar radiation suggests that it would be wise to use some type of awning or shield to prevent sunshine from reaching the window glass. Venetian

TABLE 3. DATA FROM

Test Room	Dry-Bulb Temperature, Deg Fahr									
	Time	Outdoors Near Wall	5 Ft from Floor	1 Ft from Floor	2 Ft from Ceiling	Average Outside the Room	Furred Space in Ceiling	Furred Space Beneath Floor	Air Entering	Air Leaving
East	7 A.M.	....	76.0	75.5	75.3	76.5	74.0	....	66.9	75.8
	8	....	75.0	74.5	74.5	75.8	73.5	74.6	64.9	74.5
	9	....	75.0	74.8	74.9	75.4	73.0	....	65.0	74.8
	10	....	75.0	74.4	74.5	75.0	73.0	74.6	65.6	74.7
	11	....	74.9	74.2	74.5	75.0	73.0	....	64.5	74.8
	12	....	75.1	75.0	74.9	75.3	73.0	74.8	65.0	75.1
	1 P.M.	....	75.0	74.5	74.4	75.3	73.0	....	64.9	75.0
	2	....	75.0	74.8	74.8	75.5	73.0	74.3	65.0	75.0
	3	....	74.9	74.2	74.5	75.5	73.0	....	65.0	74.7
	4	....	75.0	74.8	74.8	75.5	73.0	74.5	64.9	74.9
	5	....	74.8	74.3	73.8	75.0	73.0	....	64.9	74.8
South	7 A.M.	....	....	....	....	....	....	....	....	....
	8	....	76.5	77.0	76.5	75.3	76.0	76.2	68.0	75.5
	9	83[a]	77.0	77.2	77.2	75.4	75.5	....	66.5	75.5
	10	87.5	77.4	77.5	77.5	75.7	75.0	75.0	66.0	75.9
	11	91	77.0	77.0	77.2	75.7	74.8	....	64.5	75.4
	12	95	77.0	77.0	77.0	76.0	74.5	74.5	64.5	75.3
	1 P.M.	98	76.0	75.8	76.0	76.7	75.0	....	64.9	75.0
	2	98	75.5	75.5	76.0	75.0	75.0	74.2	65.5	74.5
	3	98.5	75.4	75.0	75.4	76.0	75.0	....	66.5	74.5
	4	97	74.5	74.1	74.8	75.3	75.0	74.5	65.0	73.5
	5	....	75.4	75.2	76.0	74.7	75.0	....	67.2	75.0
West	7 A.M.	71.8[b]	74.5	73.8	74.1	76.0	75.0	....	65.2	74.5
	8	71.7	74.9	74.2	74.7	75.3	74.2	75.4	66.3	74.8
	9	75.7	74.8	74.1	74.4	75.0	74.0	....	66.2	74.5
	10	81.8	75.1	74.5	74.8	75.0	74.0	74.8	66.6	74.8
	11	84.6	74.8	74.0	74.5	74.8	73.0	....	62.9	75.0
	12	89.0	74.9	73.9	74.5	75.0	73.0	74.0	62.0	74.1
	1 P.M.	94.3	75.2	74.0	75.2	75.5	72.8	....	61.7	74.5
	2	96.5	75.0	73.8	75.0	75.3	72.0	74.0	60.5	73.8
	3	97.9	75.3	74.5	75.2	75.8	72.0	....	61.0	74.0
	4	99.7	75.4	74.5	75.1	75.8	71.8	74.0	59.0	73.8
	5	94.4	74.8	73.8	74.5	75.0	72.0	....	62.1	74.1
North	7 A.M.	[a]	75.0	75.0	75.0	76.0	75.5	....	74.0	75.5
	8	....	75.0	75.0	75.0	75.5	75.5	....	73.0	75.0
	9	77	75.0	75.0	75.0	75.3	75.5	....	72.5	75.0
	10	81	75.5	75.0	75.0	75.3	75.5	....	72.5	75.0
	11	84	75.5	75.0	75.0	75.3	75.5	....	72.5	75.0
	12	87	75.2	75.0	75.2	75.3	75.5	....	72.2	75.0
	1 P.M.	88	75.2	75.0	75.0	75.6	75.0	....	71.2	75.0
	2	90	75.0	75.0	75.0	75.0	75.0	....	69.5	75.0
	3	92	75.2	75.0	75.0	74.5	75.0	....	70.0	75.0
	4	94	75.5	75.0	75.0	75.5	75.0	....	70.5	75.5
	5	....	75.0	75.0	75.0	75.5	75.0	....	70.5	75.0

[a] Temperature taken 6 in. from wall.
[b] Temperature taken 1 in. from wall.

blinds and other types of inside window shades do not prevent solar radiation from entering through the glass although they doubtless do reflect some of it out again. Tests made on two successive days in the latter part of September, one day with the Venetian blinds covering the whole window and with the

TEST ROOMS, SEPT. 11, 1931

Volume of Air Entering Room, cfm	Heat Removed from Room by Air, Btu per Hr	Sunshine Through Window Glass			Heat Through Glass by Conduction, Btu per Hr	Heat from Surrounding Space, Btu per Hr	Net Heat Entering Room, Btu per Hr[a]
		Normal Area, Sq Ft	Btu per Sq Ft per Hr	Btu per Hr			
...	....	30.5	88	2685	−460	−84	2368
231	2435	22.0	161	3540	−290	162	3639
204	2198	12.5	205	2560	210	−68	2929
175	1748	2.8	219	615	670	−103	1409
88	996	0	0	0	850	−29	1048
75	832	0	0	0	1110	−161	1176
87	945	0	0	0	1190	−42	1375
93	1020	0	0	0	1350	−135	1442
128	1368	0	0	0	1440	85	1752
74	812	0	0	0	1500	−47	1685
112	1218	0	0	0	1430	21	1678
...	....	16.0	88	1410	−480	....	....
229	1885	22.0	161	3540	−430	−544	2800
320	3160	26.6	205	5455	80	−921	4845
348	3780	29.3	219	6415	510	−1084	6075
338	4045	28.2	219	6175	674	−930	6153
343	4070	22.2	228	5060	950	−856	5388
355	3935	13.0	237	3080	1110	−44	4380
268	2648	4.7	223	1050	1300	−421	2163
280	2460	0	0	0	1340	74	1648
270	2521	0	0	0	1500	307	2041
159	1360	0	0	0	1460	−308	1386
...	....	0	0	0	−350	782	658
60	557	0	0	0	−280	210	156
76	688	0	0	0	210	156	592
74	665	0	0	0	660	−88	799
0	0	0	0	0	800	−119	907
108	1443	0.7	228	160	1100	−174	1312
167	2365	11.0	237	2605	1200	−116	3915
313	4615	22.7	223	5060	1320	−77	6529
373	5375	33.0	200	6600	1360	−181	8005
373	6120	31.0	183	5675	1440	−251	6868
132	1775	0	0	0	1400	−128	1469
...	....	0	...	0	−370	322	185
54	116	0	...	0	−300	251	184
54	145	0	...	0	210	108	551
54	145	0	...	0	680	−34	879
94	255	0	...	0	760	−34	959
111	335	0	...	0	1100	36	1369
132	545	0	...	0	1190	71	1494
139	834	0	...	0	1340	0	1573
48	261	0	...	0	1420	−200	1453
75	410	0	...	0	1430	0	1663
146	714	0	...	0	1360	−143	1450

[a] Heat given off by observers assumed to be 100 Btu per hour. Heat entering through wall assumed approximately 125 Btu per hour in east and west rooms and 135 Btu per hour in north and south rooms.

slats closed and the other day with the blinds up, entirely exposing the windows, showed that the rooms required as much cooling with the blinds as without them. It is possible that further tests would show that the blinds do make some difference but less than commonly supposed.

As a matter of interest to determine the absorption of solar radiation by double windows, pyrheliometer readings were taken through two panes of ¼-in. plate glass spaced 1¾-in. apart. The panes of glass were as clean as it was possible to make them. The absorption was 16.7 per cent through one pane of glass and 37.5 per cent through two panes. This indicates that double windows are of value in the summer in reducing the solar radiation entering through the windows as well as in reducing the amount of heat entering by conduction.

In all test rooms the actual amount of cooling required was less than the net calculated amount of heat entering the test rooms. In general the discrepancy was greater when the sun was shining in through the windows than at other times indicating that a large part of the error was in calculating the solar radiation delivered to the room. There are several possible explanations for this:

1. The window glass in the test rooms was not perfectly clean, whereas pyrheliometer readings taken to determine the absorption of solar radiation were made with clean glass. It is thus possible that the assumed loss in solar radiation in passing through the glass was too low and that the actual amount of heat delivered to the rooms was less than calculated.

2. During the tests the blinds were let down from the top as far as the meeting rail of the windows but were opened so that the slats lay in horizontal planes. This in general prevented the sunshine entering through the upper sash from striking the floor or walls of the room. The area of sunlight was calculated as though the blinds had not been there. It is possible that some of the heat was reflected out through the glass by them.

3. The temperature of the window glass is increased by the absorption of solar radiation and therefore the transmission of heat from the outdoor air through the glass is decreased and was probably less than calculated.

4. It is probable that part of the solar radiation striking the floor and interior walls was absorbed by them and transmitted to the spaces outside the room.

The lag in the cooling load in the south test room after the sun stopped shining through the windows was doubtless due in part to the heat being given up to the room by the floor and walls on which the sun had been shining. It will be noted from Fig. 9 that in the south room the temperature was above 75 F during the time the sun was shining in through the windows. This is because the part of the system serving this room did not deliver air of a sufficiently low temperature nor enough air to cool this room to 75 F.

The air quantities and temperatures are given in Table 3. The lowest air inlet temperature for the day in this room was 64.5 F whereas in the west room a temperature of 59 F was reached. The fan for the south end of the building ran at full speed from 9:30 A.M. to 1 P.M. but the air pressure in the supply duct was not great enough to supply sufficient air through one diffuser opening to handle the cooling load. If a temperature of 75 F had been maintained in the south room the actual cooling load would have been considerably higher.

The heat transmitted through the glass by conduction was calculated from the Weather Bureau temperatures. The first column in Table 3 indicates that the temperature varies on the different sides of the building but these temperatures were not considered sufficiently accurate to use in the calculations.

The calculated heat flow through the outside wall into the test rooms based entirely on inside and outside temperatures varied from approximately —80 Btu per hour (flow outward) at 7 A.M. to +315 Btu per hour at 4 P.M. when the outside temperature was a maximum. Such calculations assume that equilibrium has been established through the wall. This assumption is, of course, incorrect in this case because the temperatures do not remain constant long enough to establish equilibrium. In view of this, the assumption used in this paper of 125 to 135 Btu per hour through the outside wall of the test rooms based on Nicholls' heat meter readings seems reasonable. At any rate, the heat entering through the wall is small compared with that through the windows, the calculated transmission through a test room wall being approximately 16 Btu per hour per degree difference and that through the windows 75 Btu per hour per degree. Just before 4 P.M. when the solar radiation entering the west test room was at a maximum, the heat through the wall accounted for only about 2 per cent of the heat being removed from the room by the cooling air.

As shown in Fig. 5, the test rooms were about 2 ft 4 in. narrower than a full bay. Because the amount of heat entering through the wall is so small, however, compared with the total, the results shown in Table 3 and Fig. 9 may be taken as substantially correct for a full bay. While the total amount of heat removed from the test rooms may be assumed the same as that from a standard office (neglecting lights and people), a lower rate of air renewal would be required in a standard office because of its greater width and depth. In the west test room under peak conditions air had to be supplied at the rate of about 375 cfm and at a temperature of 59 F in order to maintain a room temperature of 75 F. This amount of air is equivalent to six changes per hour in the office which would occupy the entire bay.

The results obtained in the test rooms cannot be considered typical of the total load of the building because of several factors:

1. Whereas the temperature of the air inside the test rooms was kept fairly constant, the inside temperature of the building as a whole varies with the outdoor temperature, being permitted to rise as high as 79 F during the hottest weather.

2. The test rooms had no lights and practically no occupants, while the building load was increased by the removal of heat and moisture from the occupants, heat and moisture from outdoor air taken into the system and heat from electric lights, fans, pumps and miscellaneous motors. The amount of outdoor air was 10 per cent of the total amount circulated except when the outdoor wet-bulb temperature was below 60 F when the amount of fresh air was increased to 50 per cent of the total and except also for an outdoor temperature of over 92 F when the fresh air intake was often closed.

3. Two basements were air conditioned, which were unaffected by outdoor temperature or sunshine.

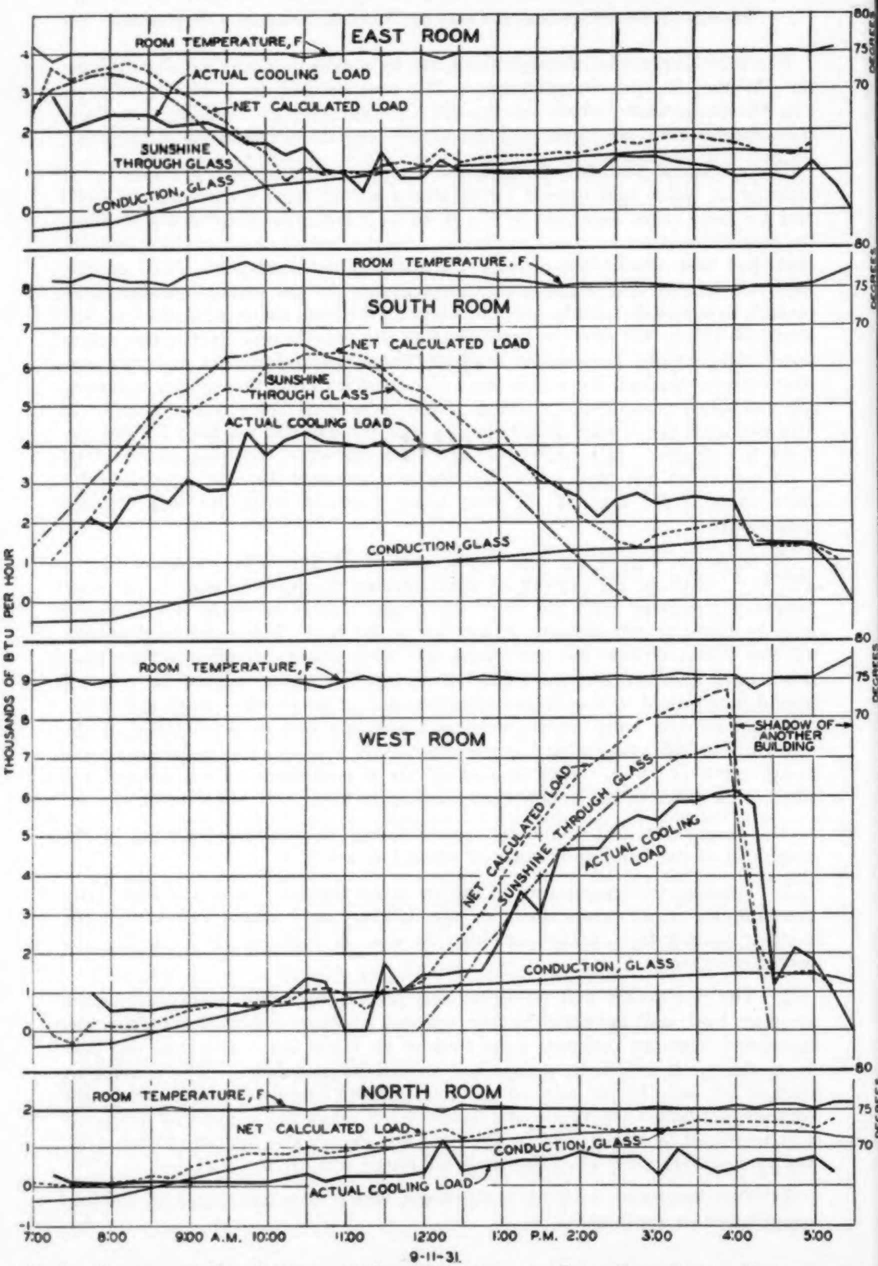


FIG. 9. CALCULATED NET AMOUNT OF HEAT ENTERING THE FOUR TEST ROOMS COMPARED WITH THE ACTUAL AMOUNT REMOVED BY COOLING

MEASUREMENT OF HEAT FLOW THROUGH WALL

The heat flow into and out of a section of the wall on the south exposure of the building was studied by means of 2 Nicholls' heat flow meters [6] placed directly opposite each other on the two surfaces of the wall. The wall at this point was made up as follows starting at the interior surface; plaster, 1¼-in.; tile, 3¾-in.; air space, 5¼-in.; tile, 3¾-in.; and brick, 9 in. The construction of the walls between the windows in the test rooms differed in that two courses of brick on the exterior were backed up solidly with brick and tile and 1 in. of plaster with no air space in the wall. The Nicholls' meter is particularly adapted to measuring the heat flow through walls of low conductivity and it is probable that the error introduced by the insulating effect of the meter plates themselves is small. There is an error, difficult to determine, caused by the different surfaces of the meter plate and the brick wall, an error which was probably greater in the case of the outside meter than the inside meter because the amount of absorption of radiant heat from the sun was increased due to the black surface of the meters. This is indicated by the fact that the temperature beneath the surface of the meter was slightly higher than the temperature of the surrounding wall surface.

Fig. 10 shows clearly how little of the heat entering the outer surface gets through the wall into the room. The sun began striking this wall at about 10:15 A.M. so that from that time on the heat absorbed by the wall was much greater than if the wall had been on the shady side of the building.

During the 24 hr from midnight to midnight on Sept. 11 the amount of heat passing through the 2 meters was as follows:

	Btu per Sq Ft
Heat entering outer surface of wall	243
Heat given up by wall to outside air	135
Net heat entering outer surface	108
Heat delivered by wall to room	29
Heat delivered by room to wall	3
Net heat delivered to room	26

This leaves 82 Btu per square foot entering the outer surface which did not appear at the inner surface. Part of this doubtless was carried off through the air space in the center of the wall and the balance doubtless went to increase the temperature of the wall. Fig. 11 shows some evidence of this as the temperature of the inner surface of the wall increased from day to day

[6] The Nicholls' heat flow meter is described in the paper, Measuring Heat Transmission in Building Structures and a Heat Transmission Meter, by P. Nicholls (A. S. H. V. E. TRANSACTIONS, Vol. 30, 1924, page 65). It consists of a thin plate of formica with a series of thermocouples on each surface. The meter is calibrated in the laboratory by determining the relation between heat flow through it and the temperature difference between the two series of thermocouples, also taking into account the surface temperature of the meter. It is fastened to the surface of the wall and the heat flow through the meter, as indicated by the thermocouple readings, is obviously equal to the heat entering or leaving the wall surface.

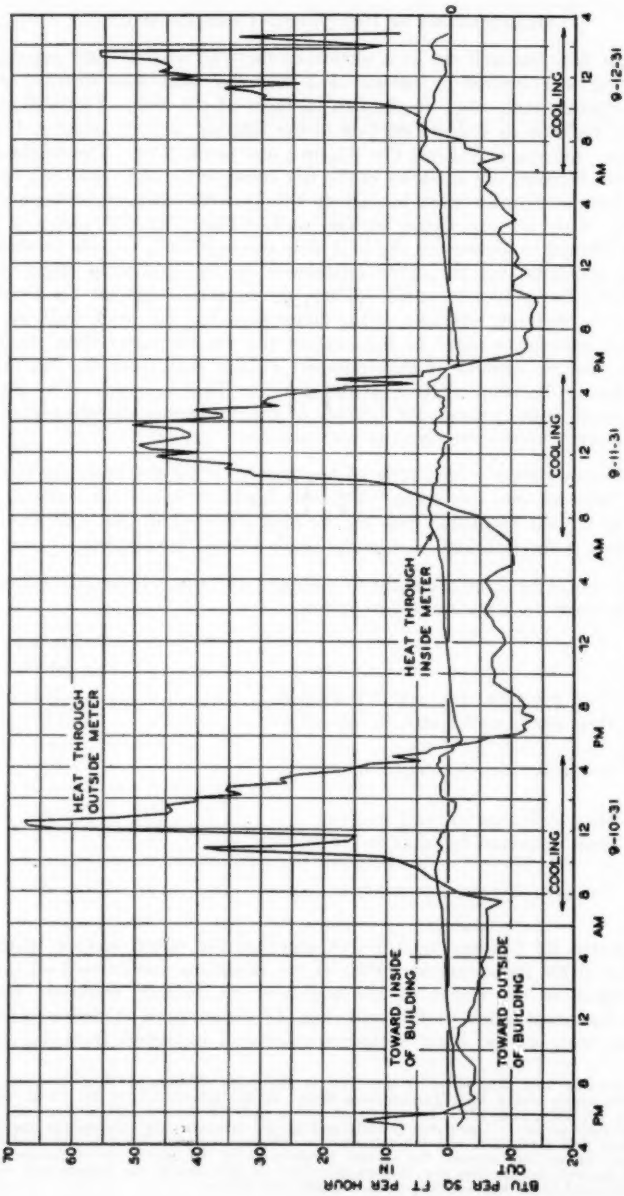


FIG. 10. HEAT FLOW THROUGH WALL AS MEASURED BY NICHOLLS' HEAT METERS

and the temperature of the outer surface was also higher each night. Calculations indicate that each square foot of wall would require 28 Btu to raise its temperature only 1 F.

A study of Figs. 10 and 11 indicates that it takes many hours for heat to pass in through the wall. The maximum absorption by the outer surface occurs from noon to 2 P.M., whereas during the night the inner surface of the wall delivers heat to the room at an increasing rate up to 7 A.M. when the air-conditioning system is started. The temperature of the inner surface of the wall rises during the night above the temperature of the air in the room indicating that this surface must be receiving heat from the interior of the wall.

Fig. 11 shows the effect of the windows on room temperature. At 5 P.M. when the air-conditioning system is shut down and when the outdoor temperature is still high the indoor temperature rises rapidly to a point above the temperature of the inner wall surface, then drops during the night to a point below the temperature of the inner surface of the wall as the outside temperature drops. Thus the outdoor temperature acting through the window glass swings the indoor temperature up and down somewhat independently of the wall temperature.

It frequently occurs that heat is flowing into a wall from both surfaces at the same time or flowing out of a wall from both surfaces at the same time. Such conditions are indicated in Fig. 10. The wall gives up heat to the outdoor air all night but after midnight when the room is cooler than the inner surface of the wall, heat is also being delivered to the room. Likewise during the day, if the indoor temperature is allowed to rise above the temperature of the inner surface, heat will flow into the wall from the room at the same time that the outer surface is absorbing heat. This occurred during the afternoon of Sept. 10 when the room temperature in the place where the heat meter was located rose to nearly 79 F before it was checked by opening the damper in the ceiling outlet. It was more difficult to control room temperature by adjusting the standard ceiling outlet than by adjusting the dampers in the test rooms. This partly accounts for the fluctuations in room temperature shown in Fig. 11.

Table 4 gives wall and floor temperatures taken in the west test room. It is rather remarkable that the wide fluctuations in the temperature of the outer part of the wall have so little effect on the temperature of the room side of the wall. The temperature 1 in. beneath the interior surface varied only 2 F in 24 hr. The temperature of the steel column is even more stable, varying only 1 F in 24 hr. In the floor at a point 1 in. beneath the surface, the 24-hr variation was ½ F. At a point 8 in. deep, the variation was 1 F. All of this shows that the temperature of the interior building structure is very stable and that if it were not for the disturbance caused by heat entering through the window glass, temperature regulation in the summer would be a simple matter in an air-conditioned building.

Another striking fact is the utter insignificance of the wall conduction in the total cooling and dehumidifying load. Fig. 12 shows the relative magnitude of the various components of the cooling and dehumidifying load as calculated for a typical bay having the exposure of the south test room at a time when

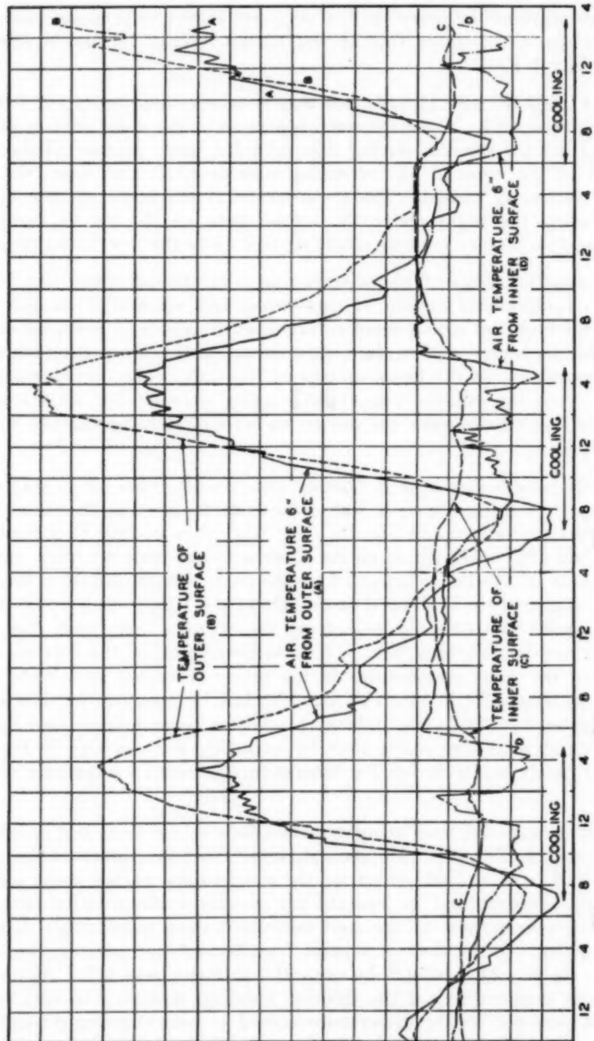


FIG. 11. WALL TEMPERATURES DURING HEAT METER TESTS

the sun radiation effect is near its maximum. The following assumptions were made:

Indoor dry bulb, 75 F; wet bulb, 62 F.
Outdoor dry bulb, 85 F; wet bulb, 72 F.
Air supply to room, 500 cfm.
Make-up, 10 per cent of air supplied.
Normal occupancy of room, 5 people.
Two 200-watt lamps, 2 more not in use.
Power for fans and pumps, average per bay for building.

Of all the factors, the heat entering through the wall is of least importance while the solar radiation through the glass is the most important factor. It should be pointed out that solar radiation through windows is not a constant and that at other times during the day the amount is less than shown here. The heat from electrical power used by fans and pumps was obtained by assuming that 90 per cent of the heat equivalent of the electricity becomes a part of the cooling load.

All of the data shown in Fig. 12 should not be taken as the average for the building because the solar radiation, number of lights, and number of people will vary in different parts of the building. Also there are other sources of heat not listed here such as infiltration, hot water piping, and miscellaneous electric motors.

Total Cooling Load of the Building

Another phase of the work was a study of the total cooling and dehumidifying load of the entire system to determine its variation with outdoor temperature and amount of sunshine. The cooling and dehumidifying load was determined by subtracting from the heat removed by the condenser, the heat equivalent of the electrical power used to drive the compressors. This gave a relatively close estimate of the load, but it cannot be considered to be entirely accurate.

The curves in Fig. 13 show the seasonal variation of cooling load, plotted with outdoor temperature and the amount of sunshine. The outdoor temperatures given are the average of the hourly readings of the U. S. Weather Bureau from midnight to midnight. Sunshine is expressed as the number of hours per day that the sun cast a distinct shadow. This does not give an accurate measure of solar radiant heat because the heat intensity varies with the time of day. On Monday evenings parts of the air-conditioned space are in use so the cooling system is operated for longer hours than on other days. In Fig. 13 the points surrounded by circles represent the actual amount of cooling on Mondays. In order to make the data comparable with those on other days, the cooling on Mondays was reduced by the number of extra hours of operation on Mondays. The points thus obtained were plotted as a part of the curve. Also on Saturday afternoons the cooling system is shut down earlier than on other days. The points plotted as crosses represent the actual amount of cooling on Saturdays. This amount was corrected by increasing it in proportion to the longer hours of operation on other days and the corrected points were plotted as a part of the curve.

During the period May to October, inclusive, the refrigerating apparatus was in operation for 1,203 hr. Of this time there were 98 hr when all three

TABLE 4. WALL AND FLOOR TEMPERATURES, WEST TEST ROOM, SEPTEMBER 11, 1931

Time	Air, 1 In. from Exterior Surface	Wall Temperatures, Deg Fahr				Air 1 In. from Interior Surface	Steel Column	Floor Temperatures, Deg Fahr	
		Exterior Surface	1 In. Beneath Exterior Surface	1 In. Beneath Interior Surface	Interior Surface			1 In. Deep	8 In. Deep
1 A.M.	80.0	81.1	83.0	79.4	78.9	77.4	76.6	75.2	74.7
2	79.2	80.4	82.4	79.3	78.9	77.3	76.6	75.2	74.8
3	77.6	79.4	81.8	79.4	78.9	77.2	76.8	75.2	74.9
4	77.4	78.6	80.7	79.3	78.9	77.2	76.8	75.2	74.9
5	74.4	76.2	79.8	79.3	78.8	77.1	76.9	75.2	75.0
6	72.5	75.7	78.5	79.2	78.8	76.8	77.0	75.4	75.0
7	71.8	74.9	77.4	79.2	78.3	75.5	77.0	75.5	75.2
8	71.7	74.4	76.6	78.7	77.8	75.2	76.6	75.2	74.8
9	75.7	76.7	77.7	78.2	77.4	75.3	76.7	75.2	74.8
10	81.8	80.7	79.7	78.6	77.8	75.5	77.0	75.3	75.0
11	84.6	82.7	81.8	78.5	77.9	75.8	77.4	75.2	74.9
12	89.0	86.8	84.6	78.3	77.5	74.7	77.3	75.1	74.8
1 P.M.	94.3	93.4	89.7	77.9	77.5	75.3	77.2	75.2	74.8
2	96.5	100.2	95.5	78.0	77.3	73.4	77.0	75.2	74.4
3	97.9	104.8	100.2	78.1	76.6	73.4	77.0	75.1	74.5
4	99.7	108.0	104.4	77.9	76.5	72.4	77.0	75.0	74.5
5	94.4	98.8	99.7	78.2	76.9	75.6	77.2	75.1	74.2
6	93.0	94.6	96.4	77.9	77.5	77.8	77.2	75.1	74.2
7	89.8	91.9	93.9	78.3	78.2	77.9	77.4	75.1	74.3
8	87.2	89.4	91.5	78.7	78.3	78.0	77.4	75.2	74.4
9	85.3	87.6	89.6	79.2	78.7	78.2	77.4	75.2	74.6
10	82.9	85.7	87.7	79.5	79.0	78.2	77.5	75.3	74.8
11	81.9	84.3	86.4	79.7	79.2	78.3	77.5	75.4	74.8
12	80.2	83.1	85.3	79.9	79.3	78.3	77.6	75.5	75.0

compressors were in use. In other words, during the hours of operation it was necessary to use 3 compressors about 8 per cent of the time.

The curves in Fig. 14 show the hourly variation of cooling load plotted with the outdoor wet- and dry-bulb temperatures and the amount of sunshine. Dry-bulb temperatures were obtained from the U. S. Weather Bureau, as well as the hourly per cent of sunshine. Wet-bulb temperatures were taken from the records of the Union Guardian Building operating engineers and were measured in an air intake.

The curves in Fig. 13 show a noticeable correlation between the total daily cooling load and the average daily outdoor temperature. This, however, does not signify that heat transmission through windows and exterior walls is the primary cooling factor, because high dry-bulb temperatures usually are ac-

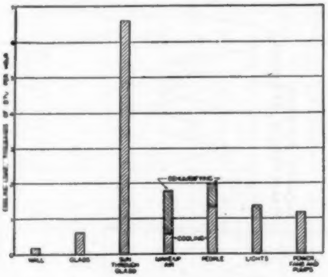


FIG. 12. DIVISION OF COOLING AND DEHUMIDIFYING LOAD, TYPICAL SOUTH BAY

companied by increased solar radiation and high wet-bulb temperatures, factors which add to the cooling load in a different manner.

It should be noticed that variations in cooling load lag slightly behind the variations in average daily outdoor temperature. The explanation of this lag can partially be understood from Fig. 14. It is apparent that the operation of the cooling system is stopped in the afternoon at a time when the outside temperature has usually reached its peak, leaving many hours of high temperature and sunshine during the remainder of the day to add heat to the building, particularly on the west side of the building. Much of this heat is dissipated outward through the windows if the outside temperature falls below the inside temperature during the night.

Two other factors operate to heat the building during the evening; one is exfiltration which takes place during a warm evening due to the vertical column of relatively cool air inside the building, 16 stories in height. The second factor is addition of heat from lights used by janitors during the evening, when no fans are being operated. During the hours after midnight heat also enters the building from the exterior walls, as shown in Fig. 10. The result is that the cooling system must be started from one to 3 hr before the offices are occupied, and cooling continued at a fairly high level during the early forenoon to care for the heat absorbed by the building and furniture.

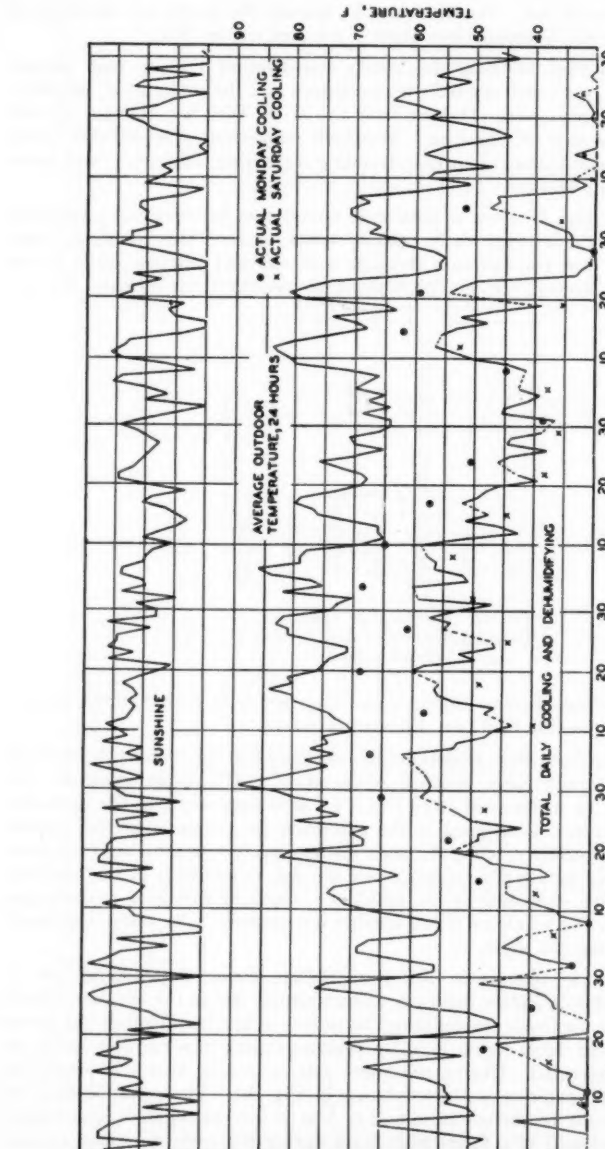


FIG. 13. TOTAL DAILY COOLING AND DEHUMIDIFYING BY THE AIR-CONDITIONING SYSTEM, SUMMER OF 1931

Sunshine, which greatly affects the cooling load of a particular room, does not affect the total load of the building to the same degree because only a portion of the building is exposed to the sun's rays at a time. Sunshine produces its greatest effect on the Union Guardian Building after 1 P.M. This is due to the orientation of the building, which is such that the east side gets the morning sun at a sharp angle, letting a relatively small amount of low intensity sunlight into the windows. The south side of the building is narrow, so that although the sunshine strikes this side squarely and brightly, only a small percentage of the total exterior surface of the building is affected. Sunlight enters the west windows after noon and a large expanse of wall and window surface is then affected by solar radiation of high intensity, except that the north half of this side is gradually shaded by a neighboring building toward the end of the afternoon.

Referring to Fig. 14, sharp increases in the total cooling load can be seen to have occurred on the afternoons of June 24 and 25, days when the sunshine was bright during the entire cooling period. The effect of heat entering the building at night, or rather the lack of heat dissipation at night, can be seen in the cooling loads of June 26 and 27. The load on June 26 between 8 A.M. and 12 noon was higher than the load on June 23 for the same period, although the dry-bulb and wet-bulb temperatures as well as the sunshine averaged somewhat lower. Comparing the same hours on June 27 with those on June 25, it can be seen that although the temperatures and sunshine are comparable, the load on June 27 is the higher, presumably because of the warmer preceding night.

Conclusions

Sunshine through the window glass is apparently the most important factor to contend with in the cooling of a room. At times it may account for as much as 75 per cent of the total cooling necessary. Because of the importance of the sunshine, cooling systems should be zoned so that the side of the building on which the sun is shining can be controlled separately from the other sides of the building. Consideration should also be given to shadows from nearby buildings which may cover part of the sunny side of a building. If buildings were provided with awnings so that the window glass were shielded from sunshine, the amount of cooling required would be reduced and there would also be less difference in the cooling requirements of different sides of the building.

In modern office buildings in the latitude of Detroit the amount of heat entering through the walls in summer is of little importance compared with that coming through the windows. It appears that the use of double windows would reduce the amount entering by direct radiation from the sun as well as that by conduction.

A study of the cooling for the entire building from Fig. 13 indicates that the total amount of cooling for any day (Mondays and days following holidays excluded) is in general proportional to the difference between the average temperature for the day and 55 F. It is affected somewhat, however, by the outdoor conditions prevailing during the previous day.

In the latitude of Detroit appropriate design figures for the heat entering the walls and windows of an office building are as follows:

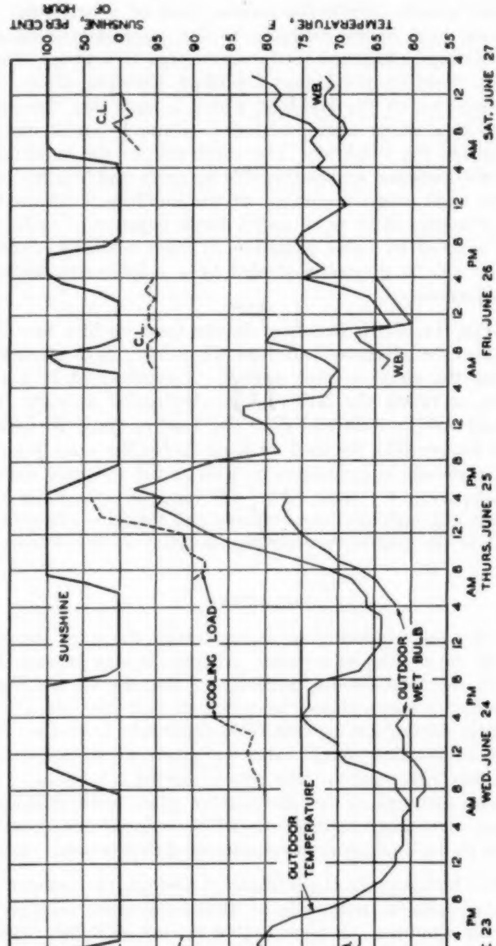


FIG. 14. VARIATION OF BUILDING COOLING LOAD WITH OUTDOOR TEMPERATURE AND SUNSHINE, JUNE 1931

Wall transmission	3 Btu per hour per square foot of wall
Glass transmission	17 Btu per hour per square foot of glass
Sunshine (east and west sides)	160 Btu per hour per square foot of glass
Sunshine (south side)	140 Btu per hour per square foot of glass

The maximum rate at which solar radiation enters south windows is less than for east and west windows because of the angle at which the sunshine passes through the glass. The glass area referred to is the area actually exposed to the sunshine. If there are overhanging ledges above the windows, these must be taken into account.

ACKNOWLEDGMENTS

The Nicholls' heat meters and the Abbot pyrheliometer used in the tests were loaned by the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. Acknowledgment is due F. C. Houghten for his helpful suggestions and comments, to the Union Guardian Trust Company which cooperated in making the tests, and to C. B. Sprenger and C. W. Signor of The Detroit Edison Company who conducted the tests and contributed many suggestions.

BIBLIOGRAPHY

See Selected Bibliography—Office Building, Heating, Ventilating and Cooling (*Heating and Ventilating*, Sept. 1931, pages 89 and 90).

DISCUSSION

L. L. SMITH AND H. L. WALTON (WRITTEN): The authors have conducted an investigation which indicates considerable differences between one factor on which the design of the installation was based and that observed in the test, *i.e.*, the sun effect. For the purpose of making a comparison, the observations and results of the test have been averaged for the south room for that period of the day covered by the hours of 10:00 A.M. to 1:00 P.M., inclusive. Considering the orientation of the building and the difference between test and solar time, this period is that during which the maximum sun effect would be had on the south exposure. The heat to be removed from the room, as observed by the test for this period and that computed under the conditions of the test by using the factors on which the design of the installation was based, is:

Type of Construction	Heat to be Removed (Btu per hour) Design	Test
Sun through glass	1980	5183
Sun on wall	474	000
Observor	100	100
Wall transmission [a]	167	135
Glass transmission [b]	925	868
Total	3646	6286
Heat to surrounding spaces	—729	—729
Total heat to be removed by air	2917	5557

[a] 79 sq. ft. @ 0.17 (87.8 — 75.4) = 167
[b] 66 sq. ft. @ 1.13 (87.8 — 75.4) = 925

87.8 deg average outside temperature by Weather Bureau.
The 100 unit for observor is carried in the design figures to make them comparative with those of the test.

The heat to be removed, computed for the conditions of 95 dry bulb 75 wet bulb outside temperatures on the basis of the factors used in the design, compares to that computed with the factors recommended by the authors as follows:

Computed on Basis of Factors Recommended.

		Btu per hr.
(1)	Sun through glass 49 sq. ft. @ 140 Btu	6860
	Wall transmission 79 sq. ft. @ 3 Btu	237
	Glass transmission 66 sq. ft. @ 17 Btu	1122
	Heat to be removed	8219
	Computed on Basis of Design Factors for 95 deg D B outside recommended conditions to 74 deg D B inside.	
(2)	Sun through glass 66 sq. ft. @ 30 Btu	1980
	Sun on wall 79 sq. ft. @ 6 Btu	474
	Wall transmission 79 sq. ft. @ 0.17 (95–74)	282
	Glass transmission 66 sq. ft. @ 1.13 (95–74)	1566
	Heat to be removed	4302

In the comparisons listed, it is found that with the items considered, the heat to be removed, based on the factors used in the design, is only approximately 53 per cent of that observed in the test or computed on the basis of the factors recommended as a result of the test. Even though such a difference would be minimized by the effect of other items entering into the total cooling load and might be reduced by compensating errors in computing other quantities, it is believed that if such a difference was actually found, the installation would be noticeably deficient in capacity.

From the method pursued and considering the quantities and factors involved, we believe that considerable dependence should be placed on the measurement of heat removed as determined from the air supply and its rise in temperature. The average heat removed per hour for the period under consideration is 3958 Btu's. The heat to be removed, determined on the basis of the design, is 27 per cent below this figure and that observed by the test is 40 per cent above.

It is evident that the difference between these results occurs almost entirely in the heating attributed to the sun effect. The effect on the total cooling load which the design contemplates for the installation is considerably minimized, however, by reason of other items entering into the makeup of this load that are not compared above. On the basis of the design for the installation, corrected for test results, the relative weights of the several items comprising the cooling load, are:

		Btu per min.	%
	Sun through glass	55.	31.
	Sun on wall	7.9	4.32
	Wall transmission	2.95	1.67
	Glass transmission	16.70	9.40
	Lights (2 watt per sq ft)	28.40	16.00
	Heat (5 people)	25.00	14.10
	Moisture (5 people)	5.68	3.21
(a)	Heat and moisture in makeup	36.12	20.30
		177.75	100.00

(a) 20 per cent makeup and 20 per cent recirculation through dehumidifier.

The total sun effect amounts to 35 per cent of the cooling load on which basis, if uneffected by compensating errors in other items, the factors used in the design would give the capacity for the installation if full dependence is put on the test observations on the heat removed, of 10 per cent below that actually required. On the same basis the factors recommended would give a capacity of 13 per cent in excess of that required. Although the actual equipment installed might have sufficient factor of safety to care for errors of 10 or 13 per cent, we believe that it would not take care of as great an error as represented in the total spread of these two

percentages, and putting dependence in the observed amount of heat removed, believe that some modification of the factors recommended would be advisable. This is further borne out by an examination of the curves in Fig. 9, which graphically compare the net calculated load and the actual cooling load. During those hours of the day when the sun effect is small it is noted that there is not a great difference between the actual and the calculated loads, but that this difference becomes considerable during that period when the sun effect is at the maximum.

The factors recommended for sun effect are understood to apply to the net glass area. It is believed that if factors are going to be established for convenient use, that it would be preferable to have these based on the masonry openings corresponding to the basis of those for heat transmission. It is also well shown by the observations submitted, that the orientation of a building can have a large effect on the total cooling load and the results obtained with any factors used would have to be applied with judgment.

We feel that the authors have made a very valuable and important research and investigation and with the observations and data submitted, that factors for calculating the cooling load can be selected with more assurance and confidence than any we have heretofore used.

H. J. MACINTIRE [7] (WRITTEN): The authors have written an excellent paper on an extremely timely subject, for which we are very grateful. They say that they can claim "no extreme degree of accuracy," and that their results are only field tests. Personally, I believe that for most design purposes tests like these are what are desired. There are, however, a few points that I would like to raise with an idea of increasing the value of the paper.

I note that in Fig. 9 comparative results are given between the net calculated load and the actual cooling load found from the amount of air admitted to the test rooms and the temperature rise of the air supplied. As the amount of heat absorbed, in the case of the south room, was 50 per cent greater than the cooling load for a period of over two hours, I feel that some explanation should be attempted. Although the authors have suggested a number of possible reasons for this lack of a complete heat balance, yet it does not seem possible that the reasons suggested can account for as much as 2000 Btu per hour. I have, therefore, checked some of the calculations.

Taking the reading at 11:00 A.M. from Table III, the refrigeration becomes

$$Q_{\text{Btu}} \text{ per hour} = \frac{60 \times 338 \times 0.2459\ (75.4 - 64.5)}{13.41} = 4060$$

Where c_p of the air = 0.2459, and specific volume of the air in cubic feet = 13.41.

The heat entering the room, also from Table III, is

Q_{Btu} per hour $= 6175 + 674 - 930 + 100 + 135 = 6154$

where these values are the radiant heat, conduction through glass, conduction through floor and ceiling, heat from observers, and heat through walls respectively.

These two values check the total given in the Table.

Under discussion of results the authors list four possible explanations of the discrepancy. I would like to ask whether any tests were made with glass under the effects of weather to see what difference there would be in the pyrheliometer readings as compared with clean glass. Also, I would like to ask whether any calculations were attempted to get a value for the solar heat reflected by the blinds, or for the effect of an increase of temperature of the window glass.

In calculating the total cooling load on the building the method used was to subtract from the heat absorbed by the condenser cooling water the equivalent in

[7] Prof. of Refrigeration, University of Illinois.

Btu of the electrical power used to drive the compressor. I believe that it would be worth while explaining the calculation more fully because it is not clear whether an allowance was made for the frictional and electrical losses in the compressor and motor. It is generally understood that the heat removed from the compressed refrigerant during the removal of superheat and liquefaction is equal to the useful refrigeration plus the work performed on the gas during compression.

MR. SANFORD: Messrs. Smith and Walton have suggested that the factors recommended in the paper for sun through glass should be modified. The factors given were determined from pyrheliometer readings. We can also arrive at the effect of the sun by taking the difference between the actual cooling loads in the west and north test rooms at the time when the sun effect is at the maximum. This gives 112 Btu per hour per square foot of glass instead of 160 from pyrheliometer readings. Further experiments are needed for a more accurate detemination.

In the design of the building a factor of 6 Btu per hour per square foot of wall was used for sun effect. Fig. 10 shows that in general the total of sun effect and wall transmission does not exceed 3 Btu per hour per square foot.

Professor Macintire asked about the readings of solar radiation. No pyrheliometer readings were taken with glass which had become dirty due to weather, nor were any calculations attempted to get a value for the solar heat reflected by the blinds. Solar radiation in increasing the temperature of the window glass would reduce the heat transmitted through the glass by conduction from the outdoor air. In the south room at 11 a.m., transmission through the glass was calculated to be 674 Btu per hour from the difference between the outdoor and indoor temperature. If solar radiation were to increase the temperature of the outer surface of the glass to 91 deg., the outdoor temperature, conduction from the outdoor air would become 0, although of course part of the heat absorbed by the glass from solar radiation would be delivered to the room. Eliminating glass transmission would account for over 30 per cent of the difference between the observed and calculated amount of heat.

In calculating the total cooling load on the building no allowance was made for the frictional and electrical losses in the compressor and motor for the reason that the measurements of condenser water flow and temperature were not accurate enough to justify refinements in the calculations. The total cooling load figures were developed only for the purpose of comparison with outdoor temperature and sunshine and should not be used for any other purpose requiring accurate values.

W. L. FLEISHER: I would like to ask whether the heat through the glass was measured with shaded glass or without. Most of the calculations, included in requirements, call for shaded glass, which would make a very material difference. I noticed that the sun load through the glass was nearly 50 per cent of the total load at a particular time. If shading cut it in half, we would come down almost to the calculated load. I don't think any cooling systems are designed to take care of solar radiation without shading.

MR. SANFORD: The tests were made with the Venetian shade blinds half way down, that is down to the meeting rails, and the slats were in a horizontal position. It is possible and probable that there was some reflection of sunlight from the blinds back out through the window glass. On the other hand, another series of tests were run at a different time, one day with the shades all the way up to the top, completely exposing the window glass, and another day with the shades completely down and the slats closed so that practically no sunshine got in. Strange to say the cooling load in the test rooms on those two days was almost identical. It was very difficult to find any real effect by lowering the shades.

The answer possibly is that the shades absorbed the heat and gave it off to the room, and whereas the occupants of the room were shielded from the bright light, still the heat effect was there just the same. There is no external shading of windows in this building.

In answer to Mr. Smith's discussion, there are several reasons why the cooling load as measured by the air did not agree with the calculated cooling load in the test rooms. One is that the windows in the test rooms were not as clean as they might have been. The window washers were kept out of there so that they wouldn't disturb the tests of the apparatus and as a result the windows were somewhat dirty. That doubtless cut down the solar radiation entering through the glass. Our calculation for solar radiation was for clean glass.

Another reason is that the solar radiation in striking the floors and walls of the room doubtless was absorbed to a considerable extent and transmitted out of the room. Thus all of the heat entering the glass in the form of solar radiation doubtless was not given to the room itself but some passed on through the limits of the room into other places. While the heat which escapes from the room in this way is not accounted for by the cooling air in the test room, this heat is of course a part of the total building cooling load.

The authors are inclined to disagree with the suggestion that the recommended values for solar radiation be reduced. Those values were taken carefully with a pyrheliometer and it is believed that they are substantially correct. They are even more conservative than figures which were given in the paper Heat Transmission as Influenced by Heat Capacity and Solar Radiation which has just been presented by Mr. Houghten. I believe in that paper 196 Btu per sq ft per hour was given for heat passing through window glass.

MR. STEVENSON: What color were the Venetian shades or blinds?

MR. SANFORD: They were buff color, the standard type of blind, I believe.

PRESIDENT CARRIER: I would like to ask Mr. Houghten whether that was vertical glass you were speaking of or was it horizontal glass?

MR. HOUGHTEN: Most of our studies were made with the glass perpendicular to the direction of the sunlight. The absorption for the double strength window glass used for that condition was about 10 per cent. By changing the angle the absorption could be increased to about 16 per cent.

PRESIDENT CARRIER: I would like to ask the author if your calculation was on the effective area at right angles to the sun's rays?

MR. SANFORD: The pyrheliometer readings were taken on an area normal to the sun's rays, but the figures of 140 and 160 Btu per sq ft per hour are for a vertical glass and are for the glass only, not for the masonry opening.

PRESIDENT CARRIER: That does not agree with Mr. Houghten's figure.

MR. SANFORD: Perhaps I am mistaken but I had understood that the 196 Btu figure given in his paper referred to a vertical glass.

PRESIDENT CARRIER: That may be part of our trouble.

MR. SANFORD: I think I can answer that. Mr. Houghten used a figure of about 300 Btu per sq ft per hour on a normal area and then I believe reduced it by about 10 per cent for glass absorption which would bring it down to 270 Btu per sq ft per hour, and then transferring to a vertical area of glass I believe he obtained the lower figure of 196 Btu. If I am in error, I would like to be corrected.

MR. HOUGHTEN: Mr. Sanford is substantially correct in his analysis of the Laboratory data. I misunderstood the first question asked concerning our results.

Upon reference to the Laboratory paper, Heat Transmission as Influenced by Heat Capacity and Solar Radiation, referred to by Mr. Sanford, our calculation is based upon the actually measured solar radiation passing through double strength window glass perpendicular to the direction of the sun's rays on September 7, 1931, or 275 Btu per sq ft per hour. If this figure is corrected for the angle of impingement it will give 260 Btu, 196 Btu and 81 Btu passing through horizontal glass,

vertical glass facing east and west, and vertical glass facing south respectively. There is a reason for our high value of 196 compared with Mr. Sanford's value of 160 Btu for the same condition; namely, for east and west walls. Our value is based upon the solar intensity in Pittsburgh on September 7, on which date it was the highest ever observed by us at the Laboratory, and the highest of any day for which we were able to examine records from the Pittsburgh Weather Bureau. As pointed out in our paper, this should be considered a maximum for the most perfect day as regards intensity of solar radiation. Perhaps, a much better average value for use is that given by Mr. Sanford. However, you obviously can expect to get the higher value occasionally.

PRESIDENT CARRIER: Apparently, our calculation on glass areas and the allowance for shading, inside shading, has been too low. There are other factors, however, which have compensated this to some extent. I would point out, as I have to the authors, that the average temperatures, for Detroit were closer to the room temperatures than in many other climates, for example, in Texas. The authors propose neglecting wall heating effect, where you have a higher average temperature. So that there is an average difference that is greater than in this test between the inside and out, you cannot neglect losses through the wall. It makes a great difference whether you have an average outside temperature, including the sunlight, of 75 deg, or an average outside temperature, including sunlight, of 85 deg, and you are maintaining between 75 and 80. It makes a great difference also upon sunlight effect. If you have a cold night and a bright sunlight day and a heavy wall, these two offset each other to a large extent and minimize the gain from sunlight. In fact, it becomes practically negligible, as shown in this paper. This cannot be taken as a generality, it is based upon practical experience.

However, I do believe that under average conditions the sunlight gain as a maximum is too high. Perhaps there is an average effect, as pointed out by the authors, through the absorption by the floors, that minimizes the maximum obtained at any one time.

MR. FLEISHER: What effect does double glass have on solar radiation and would it be possible with double glass to set the two glasses at angles which might reflect part of the solar radiation. Has this ever been attempted?

MR. SANFORD: One part of the test consisted of readings taken through double glass. We placed another window glass parallel to the glass in the wall and took readings through it with a pyrheliometer. The reduction in solar intensity through one pane of glass, as measured by the pyrheliometer, was around 17 per cent. Through two panes of glass it was about 38 per cent, so it appears that the use of double glass does have advantages from the standpoint of keeping out solar radiation as well as from the standpoint of preventing the flow of heat by ordinary transmission.

As Mr. Houghten has pointed out, the absorption can be increased somewhat by changing the angle of the glass, but the gain is not a large one. So far as known, double glass has never been set at angles for this purpose.

No. 925

A STUDY OF INTERMITTENT OPERATION OF OIL BURNERS

By L. E. SEELEY[1] (*MEMBER*) AND J. H. POWERS[2] (*NON-MEMBER*), NEW HAVEN, CONN.

This paper is the result of research conducted at Yale University in cooperation with the A. S. H. V. E. Research Laboratory and the American Oil Burner Association

INTERMITTENT operation of an oil burner refers to its occasional use in order to vary the average heat output required by heating practice. The greater the proportion of time that the burner is idle the lower will be the average heat output of the boiler in which it is installed and vice versa. The maximum heat output will be secured when the burner operates continuously and in practice this output is appreciably greater than the maximum heat loss of the structure to be heated. It is evident that a burner cannot operate or be *on* for any considerable period of time.

It is therefore a characteristic that most oil-fired heating systems operate intermittently and the question arises as to the effect of this on the economy of operation and also whether or not there is a preferred method of intermittent operation control. A comprehensive study should consider other methods of achieving the variable heat output required and their influence on economy of operation.

There are two other methods, namely the *continuous* and the *high and low*, the manual control not being considered because of its unpredictability in actual practice. The *continuous* method refers to those burners which operate steadily throughout the heating season and obtain variable heat output by automatically changing the rate of fuel burning or size of flame to suit the immediate requirements. The *high and low* method refers to a continuously operating burner which fluctuates from a low to a high flame and vice versa, the average heat required being obtained by regulating the duration of the *high* flame. The most obvious difference between these two methods and the one first mentioned is that there is a certain minimum output of heat that cannot be avoided, whereas the *on and off* burner may vary the heat output

[1] Assistant Professor of Mechanical Engineering, Sheffield Scientific School, Yale University, New Haven, Conn.
[2] Graduate student and Assistant in Mechanical Engineering, Sheffield Scientific School, Yale University, New Haven, Conn.
Presented at the 38th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Cleveland Ohio, January, 1932, by L. E. Seeley.

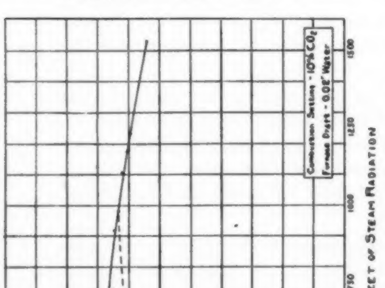


FIG. 1. CONTINUOUS AND INTERMITTENT TEST EFFICIENCY CURVES

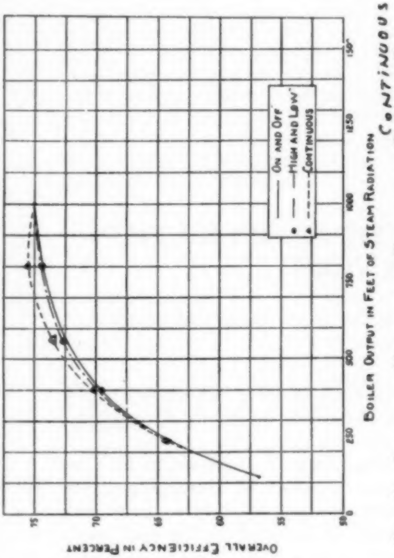


FIG. 2. ADJUSTED EFFICIENCY CURVES BY ON AND OFF, HIGH AND LOW AND INTERMITTENT METHODS

CONTINUOUS

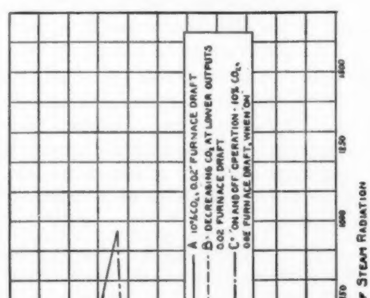


FIG. 3. EFFICIENCY CURVES OF CONTINUOUS TESTS, CONTINUOUS TYPE AND INTERMITTENT TYPE TESTS

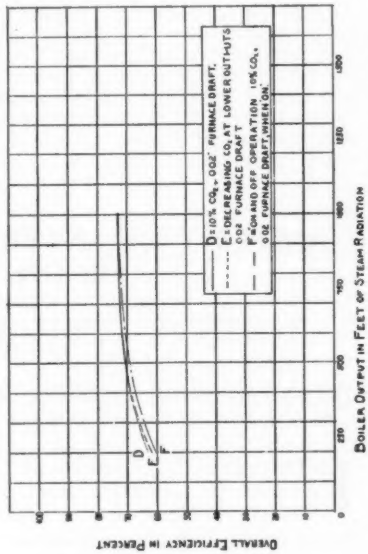


FIG. 4. EFFICIENCY CURVES OF CONTINUOUS TESTS, CONTINUOUS TYPE AND INTERMITTENT TYPE TESTS

from zero to its maximum. The study of these methods will be confined largely to matters of economy or efficiency of operation although there are undoubtedly other operating features equally worthy of study.

The A. S. H. V. E. Proposed Code for Testing Steam Heating Boilers Burning Oil Fuel [3] provides for the making of intermittent tests. This was necessary in order to obtain an idea as to what might be expected from the equipment as it is actually used. Efficiency curves obtained by the methods of heat output control in common use (*i.e. intermittent* or *on and off, continuous* and *high and low*) should indicate the economy that may be expected in actual practice.

EFFICIENCY CURVES

It was shown in a previous paper [4] that intermittent efficiency curves have a definite character although the curves obtained by continuous tests at various fuel rates do not. The continuous tests above mentioned, in distinction to the tests obtained from the continuous-type burner, are tests run at various fuel burning rates with conditions (*i.e.* rate of fuel burning, furnace draft and excess air) maintained uniform throughout each test. (See Series *A* tests [4] where the furnace draft was in practically every case kept at 0.02 in. of water and the excess air at about 50 per cent.) Fig. 1 shows two curves, one obtained by intermittent and the other by continuous tests of the Series *A* type. The extreme righthand end of the intermittent curve represents the output and efficiency when the burner is *on*. This point would naturally lie on the other curve, since it was obtained from a continuous test. The other values along the intermittent curve are obtained by operating the burner only a portion of the time during the test period.

The intermittent curves of Fig. 1 were obtained by setting the burners to operate for 30 min under definite conditions of fuel rate, draft and excess air, followed by *off* periods of different durations for different tests. Comments on this procedure indicated a desire to know what the results would have been with *on* or operating periods of other than 30 min, the feeling being that while 30 min might be a reasonable period to choose it was still arbitrary and gave no assurance that the results were typical of actual conditions. Furthermore, it did not show what was to be expected if short *on* periods were used as might be the case if steam pressure rather than thermostatic regulation were employed.

Before replying to this question it would be advisable to ascertain just what the intermittent tests with the 30-min *on* period show. All curves in Fig. 2 were, for the purpose of this paper, plotted by raising or lowering the actual curves so they would start at an output of 1,000 sq ft [5] of equivalent radiation and 75 per cent efficiency, this being analogous to the extreme righthand point of the intermittent curve of Fig. 1. The object was to superimpose the curves to ascertain whether or not there was a resemblance in shape. This would show for every combination of boiler and oil burner used to date what the intermittent efficiency curve would be if, during the *on* period, its efficiency were 75 per cent.

[3] A. S. H. V. E. TRANSACTIONS, Vol. 37, 1931.

[4] Study of Performance Characteristics of Oil Burners and Low-Pressure Heating Boilers, by L. E. Seeley and E. J. Tavanlar, A. S. H. V. E. TRANSACTIONS, Vol. 37, 1931.

[5] One square foot = 240 Btu per hour at 215 F (steam) and 70 F (air).

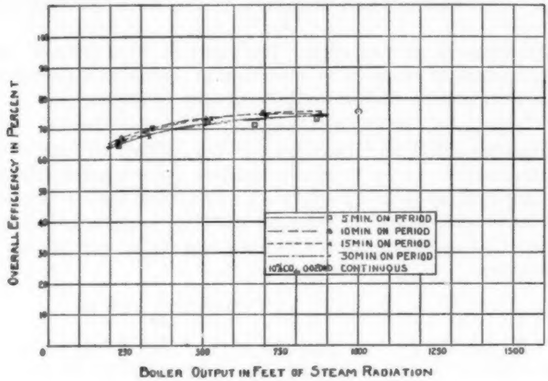


FIG. 5. INTERMITTENT EFFICIENCY CURVES *ES1* SHOWING RELATION BETWEEN BOILER OUTPUT AND OVERALL EFFICIENCY

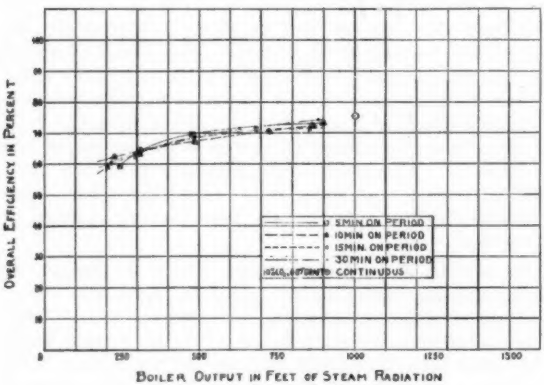


FIG. 6. INTERMITTENT EFFICIENCY CURVES *FS1* SHOWING RELATION BETWEEN BOILER OUTPUT AND OVERALL EFFICIENCY

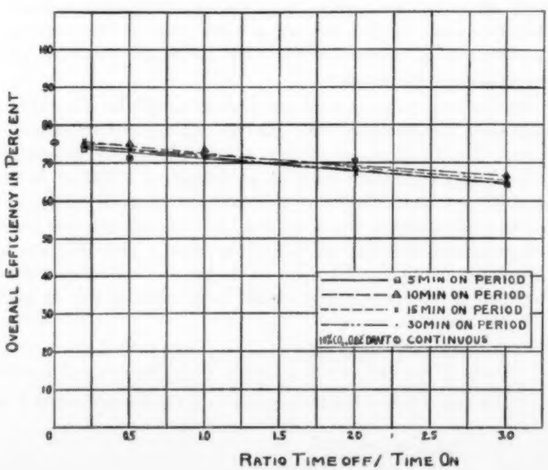


FIG. 7. INTERMITTENT EFFICIENCY CURVES *ES1* SHOWING RELATION BETWEEN OVERALL EFFICIENCY AND RATIO OF TIME OFF TO TIME ON

Fig. 2 shows an *on and off* efficiency curve. Only one curve of this type is shown because the shapes of all the curves of this type were practically the same. It can be stated with reasonable certainty that all *on and off* oil burners adjusted to the same furnace draft and same percentage of excess air which can give the same boiler output and efficiency when *on* will produce practically identical results on intermittent operation. Such combinations of oil burners and boilers should, for similar heating demands, show the same economy. It should be repeated that the sole object here is to show the trend of intermittent curves. Actual combinations may vary above or below but parallel to the curve shown.

An average curve for *continuous* burners is also shown in Fig. 2. It should be noted that the curve is about the same as the other though slightly higher. However, the conclusions relative to operating economy previously expressed are not correct for this type of burner. If two boilers were each equipped with two *continuous* burners similarly adjusted and giving the same output and efficiency at maximum fuel rate, it does not follow that identical economies would result in actual use. Heat absorption characteristics of the boiler will have an influence which is totally lacking with the *on and off* burners. Figs. 3 and 4 will illustrate the reason.

INFLUENCE OF BOILER CHARACTERISTICS ON CONTINUOUS TYPE BURNER

Fig. 3 shows that if maximum output is beyond the maximum efficiency of the boiler, the continuous burner will tend to follow the curve A for a while and then fall away at low outputs while the *on and off* curve C starts to fall away almost immediately. Fig. 4 shows that if the point of maximum output is at or near the maximum efficiency point of the boiler the continuous burner will tend to follow the curve as before and hence will drop downward in the same manner as the *on and off* curve.

Curves A and D, Figs. 3 and 4, represent the continuous tests of the series A type shown in Fig. 1. They show the heat absorbing characteristics of the boilers. Curves B and E are those obtained from a burner of the so-called *continuous* type. It is a characteristic of this type of burner to increase the percentage of excess air at the lower fuel burning rates. If the continuous type burner could automatically preserve a constant proportion of excess air at all fuel burning rates, the curves A, B and D, E would each coincide. As it is the curves B and E are each respectively influenced by the heat absorbing characteristics of the boiler which are most clearly shown when all conditions influencing performance are carefully controlled (*i.e.* curves A and D). The *on and off* curves C and F are, on the other hand, exactly the same in accordance with the findings illustrated by Fig. 2. In actual practice comparisons between *on and off* and *continuous* burners would be complicated by the fact that the burners would probably not be set for the same maximum output to take care of identical heating loads. Figs. 3 and 4 do not represent actual test curves but do represent characteristics found in tests and are merely combined to make clear the distinction that is evident in actual tests. The boiler on which the curves of Fig. 3 are based should give better results with a *continuous* burner than with an *on and off* burner over the range of outputs shown. On the other hand, there would be very little or any choice of method for

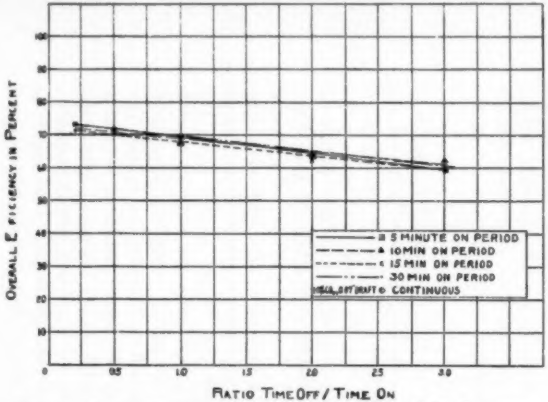


FIG. 8. INTERMITTENT EFFICIENCY CURVES *FS1* SHOWING RELATION BETWEEN OVERALL EFFICIENCY AND RATIO OF TIME OFF TO TIME ON

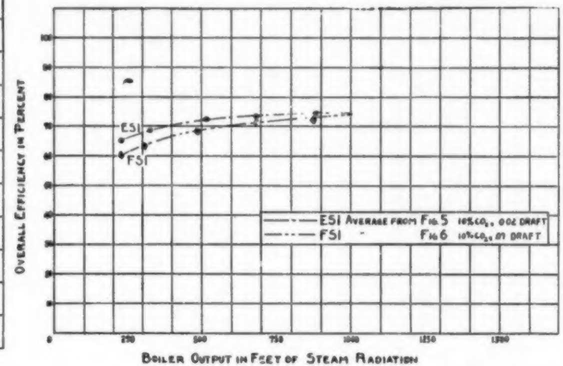


FIG. 9. AVERAGE INTERMITTENT EFFICIENCY CURVES

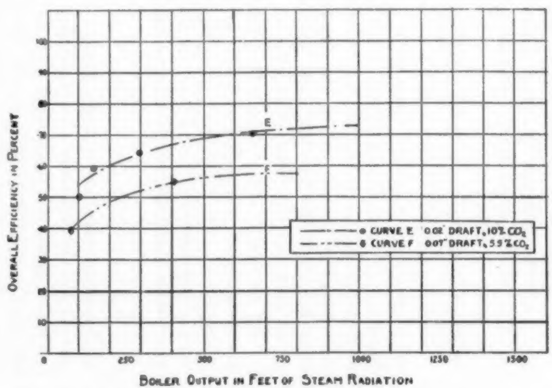


FIG. 10. INTERMITTENT EFFICIENCY CURVES SHOWING EFFECT OF FURNACE DRAFT FLUCTUATIONS

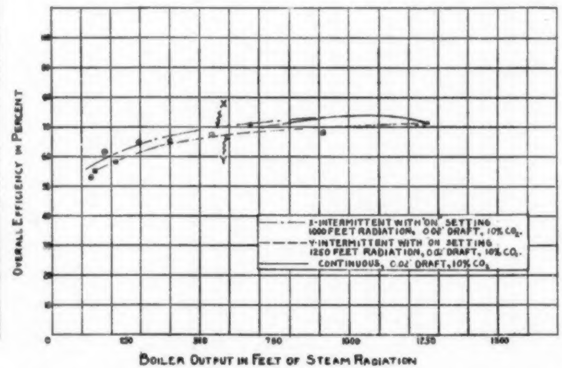


FIG. 11. INTERMITTENT EFFICIENCY CURVES AT DIFFERENT OUTPUT SETTINGS

the boiler shown in Fig. 4. Low average outputs were not obtained and comparisons should not extend beyond the limits actually shown.

Fig. 2 also shows results that may be obtained with *high and low* burners. They are practically the same as those obtained from the *off and on* types. It shows in this case that the efficiency with the *low* flame is lower than that obtained with the *high* flame. It seems reasonable to suppose that the average efficiency due to *high and low* operation will lie between the two values, never higher and never lower. Improvement in average efficiency would result from a better *low* flame operation (*i.e.* less excess air).

On Periods of Different Durations

With such a similarity shown for *on and off* efficiency curves where the *on* period used was 30 min, how will the economy be influenced by *on* periods of different duration? To answer this question, *on* periods of 5, 10, 15 and 30 min were selected and four *on and off* efficiency curves determined. *Off* periods were chosen to give the following time ratios[6]: 0.0, 0.2, 0.5, 1, 2 and 3.

Fig. 5 shows Series *ES1* tests which were obtained by setting the furnace draft at 0.02 in. of water and the combustion to show 10 per cent CO_2, while the burner was *on*. Automatic draft regulators were used. The efficiencies are not very different, all falling within the limit of probable testing error which leads to the conclusion that the 30-min *on* period tests were typical of intermittent tests in general.

Fig. 6 (*Series FS1*) shows the same result insofar as effect of timing periods is concerned. This series differs in one respect from *ES1*. The furnace draft was set at 0.07 instead of 0.02 in. of water; the CO_2 was kept at 10 per cent as before. Figs. 7 and 8 show the same efficiency curves plotted in Figs. 5 and 6 except that the ratio of the *off and on* periods is used instead of boiler output.

Effect of Furnace Draft

Comparison of Series *ES1* and *FS1* shown by average curves in Fig. 9 reveals that the average efficiency with higher drafter is not as good as *ES1* in spite of the fact that both *on* adjustments produced the same results during periods of burner operation. This suggests that for best results burners should be adjusted for proper fuel rate and combustion with as low a furnace draft as advisable for the particular installation conditions. A furnace draft of 0.02 in. of water would probably be considered too low for *on and off* operation but 0.05 in. of water should prove satisfactory in the majority of cases. The exception may be found among those burners which do not supply all the air required for combustion by means of a fan. These burners should be adjusted for proper combustion first of all and the furnace draft to supply the required air may or may not equal the recommended value. Fig. 9 explains why the regulation of draft to similar values was one of the prerequisite conditions mentioned in the discussion of Fig. 2. If this were not done it would be possible to get two combinations of boilers and burners to produce identical results while *on* but if the furnace drafts were different the economy of intermittent operation would not agree.

[6] Ratio $= \frac{\text{time off}}{\text{time on}}$

The information given in Fig. 9 should not be confused with results reported in a previous [7] study showing the effect of draft fluctuations on burner performance. Those tests were obtained by adjusting the burner as described for Series *ES1*. The furnace draft was then increased to 0.07 in. of water but in distinction to Series *FS1* no air adjustments were made to secure 10 per cent CO_2. The combustion was allowed to go where it would. Fig. 10 shows the influence that draft fluctuations may have on the intermittent operation economy. While some *on and off* burners are more affected than others in this respect, it may be concluded that once the proper draft setting is made it should be maintained as constant as possible.

THE SETTING OF AN INTERMITTENT BURNER IN RELATION TO BOILER EFFICIENCY

May one capacity setting for a combination of boiler and burner be better than any other from the standpoint of economy? Fig. 11 shows two intermittent efficiency curves. In one case the *on* setting was at a capacity of 1,265 sq ft of radiation and 72 per cent efficiency and in the other 1,000 sq ft of radiation and 74 per cent efficiency. It should be noted that the intermittent curve *x* is the higher of the two and its *on* period efficiency is higher. This would be expected from a study of Fig. 2. The best setting for economy in any case, therefore, is that operating capacity which gives the highest efficiency of the boiler in question. If this type of setting is properly proportioned to the heating load requirements of an actual building, the highest fuel economy should result. It might be again noted that Fig. 3 suggests the conditions best suited to a *continuous* type burner and Fig. 2 suggests that the *high and low* type should be treated like the *on and off* type.

It must be remembered that all tests referred to were made on the equivalent of an ordinary low-pressure steam system. It is not at all certain that a warm air, hot water or vapor system would show the same results because the heat could probably continue to flow into the system for a longer time during the *off* period and the loss of heat up the chimney might be reduced. There is no definite proof of this but plans are being made to investigate the question. Certain types of systems might prove more desirable than others from the restricted viewpoint of fuel economy to which this study has been confined.

CONCLUSIONS

The following conclusions seem to be justified from the information at hand:

1. The intermittent efficiency curve for *on and off* and its equivalent for other types of burners is the true indicator of fuel economy to be expected in actual practice.

2. All intermittent curves or equivalent have substantially the same characteristic shape for similarly adjusted burners.

3. Any number of boiler and intermittent burner combinations which have the same capacity, efficiency, furnace draft and excess air or combustion setting while *on* will produce substantially identical results.

[7] Study of Performance Characteristics of Oil Burners and Low-Pressure Heating Boilers, by L. E. Seeley and E. J. Tavanlar, A. S. H. V. E. TRANSACTIONS, Vol. 37, 1931.

4. Continuous-type burners are influenced by the heat absorption efficiency of the boiler over the whole range of output.

5. *High and low* burners tend to follow the *on and off* type of efficiency curve. The curve should be confined between low and high limits of efficiency actually obtained by the *low* and the *high* flames respectively.

6. The lower the furnace draft setting and the more uniformly it is maintained, all other things being equal, the better the fuel economy.

7. Various *on* period timings seem to have practically no effect on the intermittent efficiencies.

8. The best economy on intermittent operation for any combination is an *on* setting which produces the highest efficiency.

9. It is possible that certain types of heating systems may be better adapted for economical intermittent operation than others but this is not definitely known.

DISCUSSION

W. H. SEVERNS [8] (WRITTEN): Some of the statements in the paper appear to be contradictory. One statement is "that intermittent efficiency curves have a definite character although the curves obtained by continuous tests at various fuel rates do not." Reference is then made to Fig. 1 which shows two curves, of which one is for continuous burner operation and the other for intermittent operation for 30-min periods. The curves are ostensibly for the same boiler and burner under identical operating conditions as to fuel, draft, and burner adjustment.

The two curves intersect at a point representing a capacity of 980 sq ft of equivalent direct steam radiation and an overall efficiency of 73 per cent. When capacities greater than 980 sq ft, equivalent direct radiation, were obtained the burner operation was continuous and the overall efficiency decreased as the capacity was increased. When the capacities developed were less than 980 sq ft, equivalent direct radiation, continuous operation showed for the most part, as far as the investigation was carried, appreciably better overall efficiencies than were obtained for the same developed capacities with intermittent operation of the burner.

Reference is then made to Fig. 2 which is an attempt to translate to a single curve sheet the performance characteristics of a burner and boiler with continuous, high and low, and intermittent operation of the burner. The curves are so shifted on the sheet that they intersect at a common point of 1000 sq ft capacity and 75 per cent overall efficiency instead of the 980 sq ft capacity and 73 per cent overall efficiency of Fig. 1. The paper states that the actual operating efficiencies of Fig. 2 were not in all cases those given and that the efficiency percentages are to be ignored. Apparently the purpose was to superimpose the three curves in order to compare the general shape of each with the other two.

Examination of Figs. 1 and 2 indicates that the curves for intermittent, high and low, or continuous operation are all definite, in spite of the statement that continuous operation curves do not have a definite character. The curves of Figs. 1 and 2 prove that continuous operation is more efficient. The curves of Fig. 2 as plotted give a misleading impression, in that they seem to indicate that all three methods of operation are about equally efficient and that the more general practice of control by intermittent operation is entirely satisfactory. Had the curves been plotted using the actual efficiencies obtained their same general shapes would have been apparent, and the variation of efficiencies secured by the three different methods of operation

[8] Professor of Mechanical Engineering, University of Illinois.

would have been more noticeable. With the curves for the three conditions of operation plotted in the usual manner, it might be possible to prove that intermittent operation of a burner is decidedly uneconomical.

DR. C. W. BRABBÉE: The low efficiencies on reduced loads are certainly a very serious thing. These tests bring us from certain assumed conditions to actual practical conditions. Even if boilers have 85 per cent efficiency (and there are some of them) in continuous operation, they do not have this efficiency in intermittent operation. Knowing that continuous operation in practice is actually obtained only on a few days, whereas in spring and fall the intermittent operation is the rule, we can see the magnitude of this influence. Further, it is to be considered that people often run oil burners in summer for hot-water supply. Also by using the burner humidity in the cellar in summertime is avoided, which is beneficial to the installation.

In summer where we may have periods of 10 min *on* and an hour or more *off*, those decreases in efficiency mean something though we must not forget that if a boiler runs 10 min *on* and an hour *off* it does not mean very much in expense as the boiler with the low efficiency is running only for a short period.

Knowing now that the decrease is quite severe I think it is proper to consider why that is so, and there are three explanations. The first is that the efficiency of combustion changes. The second is that there is a heat loss through the chimney and the third is there is a heat loss by the firebrick lining of the boiler. In this respect I would try to bring out a few points not as a criticism but insofar as I think this work should be continued and that we try to contribute something to the extension of the investigation.

The efficiency of combustion is taken care of by those curves, but two other influences have not been touched. Suppose we have two boilers; the one is a straight up-draft boiler, the other is a boiler with up-draft and a low down-draft. Don't you think that this boiler with the straight up-draft will lose, when *off*, more heat than the boiler with a deep down-draft? I think investigations in this line would be beneficial.

If we have a boiler with up-draft, we need only a little fan pressure because the boiler depends on its own draft created by its operation. If, however, we have a boiler with a deep down-draft, we need a powerful fan to overcome this resistance of the down-draft, but the minute the boiler stops operation the down-draft flue will put a stop to some of the losses.

Therefore the question would be, is there any influence on these curves if we investigate up-draft boilers or down-draft boilers, the one with only a little power and push in the fan, the other with a big push which stops the minute the burner stops?

The losses through the stack have also something to do with the fact if the boiler has fire-brick lining or not. Suppose we have one boiler which has no fire-brick at all. When the burner stops there is only a little heat to lose. But let us think of another boiler which is packed with fire-brick and which is tremendously hot when the boiler operates. Then when the boiler stops all this hot brick work loses heat through the chimney independent of whether we have a draft regulator or not. Therefore the boiler design and also the fact whether the boiler is lined or not lined, and how much lining, has some influence on the efficiency of interrupted operation.

It is certain that if a boiler is very well insulated the losses through insulation are small. We know if we run a boiler with 100 per cent load, its insulation loss may be 3 per cent, but if we run this boiler with 25 per cent load, the insulation loss may be 12 per cent. Therefore the question comes up how much does this insulation influence the falling off of the efficiency curve? Is it advisable and to what extent to insulate oil burners more than any other boilers?

The question is quite complicated and I think it is necessary that investigations continue.

H. M. Hart: I do not agree with the author on one point: that is that both the intermittent type and the continuous type of burner should be set to operate at maximum output at the point of highest efficiency of the boiler.

The statement is true insofar as the intermittent type of burner is concerned but not so with the continuous type. The continuous type of burner can be adjusted to give its highest combustion efficiency at almost any point in its range of output.

Therefore my opinion is that it should be adjusted to give its highest combustion efficiency at the normal load or the load the boiler will be called upon to carry for the greatest number of hours during a heating season.

Likewise if the boiler selected is to be fired with an intermittent type of burner it should be able to carry the maximum load at its highest point of efficiency while the boiler that is to be fired with the continuous type of burner should be able to carry the normal load at its highest point of efficiency while at the same time being able to carry the maximum load regardless of efficiency.

So it seems to me that a smaller boiler can be used with a properly adjusted continuous type of burner than could satisfactorily be used with the intermittent type.

It also seems to me that the continuous type burner would be more economical in cost of operation because it would operate at the highest point of efficiency of boiler and burner at normal load while the intermittent burner would only produce the highest efficiency at maximum load which only occurs for a small fraction of time during a heating season.

Mr. Seeley's tests clearly indicate that the longer the *off* periods the lower the efficiency with intermittent burners.

F. W. Hvoslef: I have been very much interested in this paper because it is a subject with which I have at one time been intimately associated and I have some points that I would like to mention in which I disagree with the author.

It is quite true that if you adjust your firebox draft to two-hundredths of an inch, you probably get the ideal conditions for operation in a laboratory, but two-hundredths of an inch of draft is too low for practical conditions in the field. The oil burner dealer has to install his burner some time in the summer or winter, whenever it is sold, and his burner must operate in September with outdoor temperatures of 60 deg or in January with outdoor temperatures of 20 below zero. It must be able to start when the stack is warm and it must be able to start when the stack is cold, and with only two-hundredths of an inch for the established draft in the firebox, you will get a bad smelling start in most cases when you start with a cold boiler.

In a laboratory it is entirely possible to show that a high-low burner or a regulated flame burner or an intermittently operated burner may under optimum conditions operate with equal efficiencies. In doing so we lose sight of the fact that the period of intermittent operation is not today nor tomorrow, for a few hours, or 30 min *on* and an hour *off*, or whatever it may be; but it is from the 15th of September to the 1st of June through eight months, and we must bear in mind that with a low flame, high-low burner, or with a burner which is operating with a continuous pilot, as some of them are called, you must have a certain minimum rate of combustion all the time in order to keep the burner going.

There are hundreds of hours throughout the heating season when you are burning oil with no efficiency at all, with no output, just to keep going. Certain figures that I have drawn up at one time indicated that with certain types of high-low burners or continuous burners, you would burn as many as two thousand gallons

of oil a year in an ordinary residence to keep the burner going and getting no good out of it at all.

There is one other statement that I wish to make, which perhaps is not exactly in place at this time, and that is that my own studies indicate that oil-burning boilers should be smaller boilers than where boilers are selected for burning coal.

PERCY NICHOLLS: I commend Professor Seeley on his method of attack. His conclusions may be debatable at times, but the method of attack in presenting information of this type which shows results under controlled and known conditions, should be far more valuable to the trade than would be results of tests conducted under conditions in which the operation of dampers or other controls were accidental. Stating the information under the controlled conditions, gives each one the opportunity to analyze and apply the data according to one's belief, and to reduce the results to those conditions which the chooser believes will best suit his equipment and his method of installation.

The opinion is expressed that it would have been worthwhile to carry the investigation one step further and to have measured the air, allotting it where possible to its modes of entrance.

MR. HART: The previous speaker left the impression that a continuous operated burner might be construed as being more extravagant in fuel consumption. There are two things we have to consider in the application of the oil burner. One is the oil burner itself, its ability to burn oil efficiently; the other is the application of the oil burner to the requirements of a heating system. I think our troubles from the heating contractor's viewpoint have been the lack of correlation between efficient combustion and efficient application. My experience has been the reverse on fuel consumption with an intermittent type of burner versus continuous operation, regardless of the fact that there are days in the heating season when the house might become overheated with the burner on at the low point.

However, I might point out that on those mild days in the wintertime it is sometimes desirable to open a window and let the nice mild air into the house. So that I don't think it is a total loss. In actual operating conditions my observation has been that the continuous oil burner showed very much better overall efficiency, in fuel consumption and heating satisfaction during the heating season.

PRESIDENT CARRIER: Mr. Hart has put his finger on an important point, namely that there is a necessary correlation between burner and boiler and requirements. Every one knows that a heating plant of small size cannot be engineered as a power plant would be engineered or operated as a power plant would be operated. I will not go into detail. I think a simple analogy will express my meaning, although I may be treading on somewhat dangerous ground.

If you are going to possess an automobile, would you buy the chassis and running gear in one place and the engine at your own selection in another and expect to get either performance or first cost, as you would if the whole plant was engineered at the factory as a whole?

PROF. G. L. LARSON: So far in this discussion, nothing has been said about the amount of brick-work or shape of combustion space used in this study. The size and shape of the combustion space has much to do with the efficiency of the set-up as a whole, and I feel that Professor Seeley's paper would be more valuable if it included a detailed description of the combustion spaces used.

I would like to ask Professor Seeley if any consideration has been given to this phase of the study.

W. J. KING: It just occurred to me in connection with the remark that Dr. Brabbée made regarding the possible losses in the stack due to the draft during the

off-period, in the case of very hot refractories it might be interesting to segregate that loss from the total and see how much it actually amounted to and whether it was worth any attempt to reduce or eliminate it. For example, it seems it might not be out of reason to arrange some sort of a little damper or control which would shut off the draft when the flame was not working. If there was any draft through the boiler which carried away a considerable amount of heat it might be reduced by cutting it off when it was not operating.

HOMER ADDAMS: I would like to get up to reflect my appreciation for the fine work reported by the paper. It points toward what was probably a small school of boiler designers who had a different idea than the large school which designs particularly for domestic use and it seems that there may have to be modifications to meet the demands of domestic service. I think this is a fine presentation and our Society owes a great debt for this fine work.

PROF. SEELEY: I want to clear up the point that is reflected in Professor Severns' remarks and also Mr. Hart's. I might say that Dr. Brabbée's interest in the low efficiency obtained by intermittent operation is worth noting. Probably the information would be even better and more complete if we had ratios of *time on* to *time off* that carried our average output very much lower. I would point out that wasn't done merely for a practical reason. The cost in time of testing amounts to a good deal and you think of those things carefully when your appropriations are fixed. It does take quite a little time to make such tests.

It will be noted that curve *A* isn't a test made with *continuous* burners and I try to point out that fact but it is the kind of test that one would attain if he had settings at various points where the draft and combustion were the same, and being rigidly maintained throughout the entire range of operation. Curve *A* was not obtained by any of the three commercial methods of obtaining variable output (*i.e., on* and *off, high* and *low,* and *continuous*). It merely shows what might happen if a new adjustment were made to certain standards of performance each time a change of output was required—not a practical procedure but one which shows another order of results and makes comparisons of more value. That was the continuous test first mentioned and I said it was analogous to the kind of test you get in fuel-burning boilers operating at different outputs steadily, but the curve obtained by the *continuous-operating* burner, was one that could not really under all conditions, that is over the whole range of operations, maintain the combustion as it was in the case of curve *A*. The increase of excess air would be greater than that attained for curve *A* and consequently you would get that falling off in efficiency.

Mr. Hart points out that the continuous burner might conceivably be better. I think I mentioned that it might be conceivably better and that where I specified the most desirable setting I was referring only to the intermittent *on and off* type of burner where the desirable setting at maximum efficiency would give the best overall operating result; that the heat-absorbing characteristic of any particular boiler throughout its entire range would affect the average performance of the continuous type of burner and this shows what I mean again because the continuous type of burner actually tries to follow that curve and then sooner or later falls away.

The criticism of Professor Severns, I think, was due to a misunderstanding, perhaps not understanding that this curve *A* was a special one merely put in to show what could be obtained when testing according to the code of the Society, and running continuously at various points.

The effect of firebrick, I don't think, will show up quite so well unless we carry these tests farther than we already have, and study more types than we used for intermittent operation.

Mr. Hvoslef said that two-hundredths of an inch of draft was too low, though ideal for a laboratory, a fact that is mentioned in the paper. We would prob-

ably have to go in the neighborhood of five-hundredths of an inch for satisfactory service.

I won't say anything about the size of the boiler. I think it is the performance that determines the best application and whether the boiler should be made larger or smaller depends upon its performance. It was contemplated also to run tests with the draft entirely shut off so that no air could flow through the boiler and up the stack and thus carry away that heat, but time has not allowed us to do that. I am sure it will make a difference.

So far as difference of design is concerned, whether boilers may be down-draft or straight up-draft, if the furnace is kept the same, whatever openings there are in the boiler for the supply of air, those areas do remain the same and if the draft is kept at two-hundredths of an inch of draft it will bring in a definite amount of air and that air will flow through the boiler and it seems to me that in some respects that would tend to show the same condition. The only question is, would the draft stay the same in various types and that is an open question. Boilers with high frictional resistance would probably not stay the same. That is, the draft regulator might absolutely close in an attempt to maintain constant furnace draft and then the furnace draft might continue to go down, and thus change the rate of flow of air and thus change that loss up the stack.

We shouldn't feel quite so bad about this low efficiency; in some respects any equipment operating intermittently is going to give similar results. We talk about the operation of gas boilers, giving high efficiencies and we know that combinations of oil burners and boilers will do the same thing, but they are both subject to exactly the same condition on intermittent operation. They are both subject to those losses and the paper I mentioned, in the bulletin of Purdue University shows that very clearly. It is unfortunate but there seems to be no way of getting around it entirely because you can't just arbitrarily cut out certain losses that are inherent in the heating assembly.

I also pointed out the fact in considering the difference in types that either a high-low or a continuous operating burner did have a minimum rate and that rate was required whether or not the heating load justified it. I pointed that out as a difference between the two characteristics and I do think that the big differences in them must be found at the very low operating points which we have not covered as widely as we might.

No. 926

ROOM WARMING BY RADIATION

By Arthur H. Barker,[1] London, England

MEMBER

IN ENGLAND radiant heating is regarded by many as superior to convection heating for 5 principal reasons:

1. It is more comfortable
2. It is hygienically superior
3. It reduces the fuel consumption
4. The heating surface is either invisible or inconspicuous
5. Comfortable conditions are rapidly reached

Room warming by low temperature radiation has not been extensively adopted in America although the principle appears to be equally as applicable to American conditions as to the much milder climate in England, of course, with suitable modifications of practice. Much more severe cold spells have to be provided against and much lower humidities generally prevail in America than in England, so that the capacity of the apparatus must be greater.

The valuable researches conducted by the American Society of Heating and Ventilating Engineers have shown the relation between the humidity and the temperature necessary for comfort. It proved that where the method of heating by convection is adopted the air temperature must be maintained generally higher for an equal degree of comfort. Similar researches are needed on the extent to which a rise of radiant temperature will allow of a reduction in air temperature.

The principal feature of radiant heating is that it raises the air temperature *less* than other methods. It might therefore appear that in this respect it is less inherently suitable for a very cold than for a mild climate. This, however, will be seen to be a fallacy when the true principles of the method are considered.

Another reason may perhaps be that the fundamental principles are not easy to understand or to grasp, so much so that many people find difficulty in believing that there is any reality in radiant heating. It is one of the objects

[1] Consulting Engineer.

Presented at the 38th Annual Meeting of the American Society of Heating and Ventilating Engineers, Cleveland, Ohio, January, 1932.

of this paper to present these principles in such a way that a person interested can not only appreciate this reality but make the necessary calculations from first principles.

Possibly one reason the development has taken place chiefly in England has been that the principle was first introduced into that country by the author, and it was there adopted by a very enterprising and able firm of heating engineers who devoted a large part of their resources to its development, and in its initial stages took some rather terrifying risks. On a small scale or in a laboratory the real difficulties that arise in a large installation are never revealed. No new system can be tried out effectively, except in several full-sized installations under normal operating conditions, and the essential defects may possibly not be revealed for many years after the plant has been in service.

Thus, in the case of the panel system of radiant heating, as originally introduced, the pipes were embedded permanently and immovably in concrete, and pipes so installed are liable after a period of use, to some dangers. These include the following, which might make the use of the whole apparatus impossible or call for extensive alterations or repairs:

1. Fracture from unequal expansion.
2. Corrosion, either internal or external.
3. Stoppage of the pipes.
4. Settlement of the structure to which most buildings are liable which might cause airlock.
5. The heat from the embedded pipes might cause extensive cracking of the plaster.

The contractor supplying such an installation has to take these very formidable risks. If he has conducted experiments in several large installations he might easily be ruined, if serious defects were to develop in all of them.

PHYSIOLOGICAL CONSIDERATIONS

The opinion has been commonly expressed that heat comfort depends almost entirely on the heat generated in the body being naturally dispersed from the surface when maintained at its normal temperature, so that the body temperature remains constant. The rate of heat production depends on the degree of muscular activity at the time, and the rate of dissipation on the surrounding conditions. Thus there are different rates of necessary heat loss from the body, according to the state of activity. Within certain limits nature provides for the variations by certain thermostatic adjustments of the body mechanisms.

Heat is lost from any warm body partly by radiation and partly by convection. The amount lost by convection in still air at a constant temperature depends solely on the temperature of the surrounding air, which would be shown by a thermometer suspended in the air with its bulb shielded from all radiation.

That lost by radiation depends not on the air at all but on the mean temperature of the surrounding surfaces, that is, on what the author has called the *mean radiant* temperature which would be shown by the reading of a thermometer suspended in the middle of a room, if all the air in the room could be abstracted so that the temperature of the surrounding air had no effect on the

thermometer bulb. The technics of radiant heating require that these two considerations be dealt with separately.

When the *radiant* and the *air* temperature are the same, that is called the (British) effective temperature [2] which determines the *total* rate of heat loss by radiation and convection combined from any body at a temperature higher than that of the air. Fig. 1 shows the relation between the British effective temperature and the rate of heat loss from the body as determined by Mr.

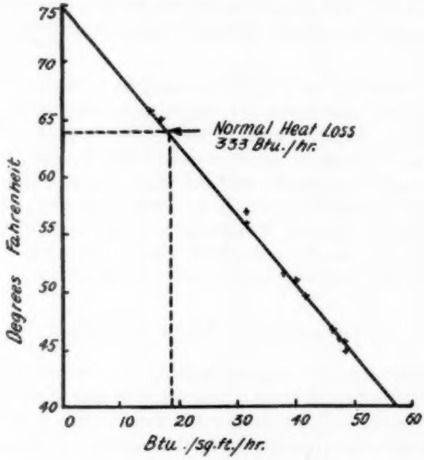


Fig. 1. Relation Between the (British) Effective Temperature and the Rate of Heat Loss from the Body

Dufton of the British Building Research Station.[3] The average surface temperature of the body including the clothes is taken as 75 F.

Thus in every warmed room there are three temperatures to consider. Generally the *radiant*, the *effective* and the *air* temperatures are all different, the *effective* being always between the other two. In a room heated by radiation the radiant is always higher than the air temperature and when heated by convection the air is always higher than the radiant temperature.

It has been in the past commonly believed or assumed that the degree of comfort depends solely on the rate of heat lost from the body irrespective of the method by which it is emitted. It is now generally accepted in England that there is a substantial difference in the feeling of comfort at the same total rate according to the way in which heat is abstracted. The higher the radiant and the lower the air temperature, the greater is the feeling of freshness for the same rate of heat loss.

[2] The (*British*) *effective temperature* is not the same as *effective temperature* as defined by the Society.
[3] Dufton: The Effective Temperature of a Warmed Room (*Philosophical Magazine*, ix, p. 858, 1930).

Thus, if the walls of a room were uniformly at 75 F, no heat would be lost by radiation.[4] In order to maintain the body comfort the air temperature would have to be about 44 F so that the whole of the heat required to be abstracted from the body at 75 F should be removed by convection currents. Similarly, if the air of the room were at 75 F, no heat would be removed by convection and the surrounding walls would have to be at an average of about 50 F in order that the whole of the surplus body heat might be dissipated by radiation alone. The former condition would produce a feeling of warmth and freshness. The latter would feel equally warm but stuffy. The thermometer reading in the two cases would be widely different though the feeling of warmth would be the same.

The feeling of comfort in a room warmed by radiation is greater, the air feels fresher, and the conditions are often more desirable than with a room heated by convection. For example, in one of the large English public schools, Epsom College, in which there are two rooms side by side, one being a lecture room warmed by radiation panels, and the other a preparation room (in which the schoolboys study) which is warmed by convection, both of which are heated to the same (British) effective temperature, it is found that the boys always crowd into the lecture room to do their studies, because it makes them feel brisker and less sleepy, and that is precisely the difference in effect.

OPERATION OF RADIANT SYSTEM

The practical problem of radiant heating has as its primary object the maintenance of the proper radiant temperature rather than the air temperature of a room. A progressive rise in the air temperature follows as a secondary effect. With a convective heating unit, the heating of the air is the primary effect, and the raising of the radiant temperature is a gradual process due to the warming of the walls by slow convection of the heated air circulating over the wall surfaces. Probably one of the principal reasons why air temperatures in America are so much higher than in England is to compensate for the lower temperature of the walls.

Radiant heat from a panel travels over the room instantaneously. All the heat delivered by the panel as radiant heat is at once distributed principally in the center of the room and absorbed by the floor, ceiling and walls. As the reflecting power of such surfaces never exceeds from 5 to 10 per cent, it follows that 90 or 95 per cent of the radiant heat impinging on the floor and walls is at once absorbed and this heat gradually raises the temperature of these surfaces and is slowly conducted through the walls. The surfaces of the walls therefore assume certain ultimate temperatures depending on their construction and exposure. The difficult part of the problem is to determine what these temperatures are. In practice they can only be determined by experience in connection with the particular climate concerned, as the necessary calculations would be far too involved for practical use.

The heat falling on carpets and furniture raises their temperature much more rapidly. This causes the warming of the surrounding air. The surfaces of the furniture near the panels will be found perceptibly warm.

---

[4] See Panel Warming, by L. J. Fowler (A. S. H. V. E. TRANSACTIONS, Vol. 36, 1930).

Table 1. Total Radiation per Square Foot

Temp.	Total Black Body Radiation Btu per Sq Ft per Hour	Total Black Body Radiativity 0.9	Total Black Body Radiativity 0.8	Temp.	Total Black Body Radiation Btu per Sq Ft per Hour	Total Black Body Radiativity 0.9	Total Black Body Radiativity 0.8
30	96.2	86.6	77	71	132.2	120.1	106
35	100.2	90	80	72	133.1	121.0	107
40	104.2	94	83.5	73	134.0	121.8	107.8
45	108.6	97.7	87	74	135.3	122.2	108.5
46	109.4	98.4	87.3	75	136.6	123	109
47	110.3	99.3	88.3	80	142	127.5	113.5
48	111.2	100.2	89	85	147.5	132.5	118
49	112.0	101	89.6	90	153	137.5	122
50	112.9	101.5	90.3	100	164.3	147	131.5
51	113.8	102.3	91	110	176.0	158	141
52	114.7	102.9	91.7	120	188.7	169.5	151
53	115.6	103.7	92.5	130	203.6	183	163
54	116.5	104.6	93.3	140	216.2	195	173
55	117.4	105.5	94	150	229.8	206	183
56	118.3	106.3	94.7	160	243.3	219	194
57	119.2	107.1	95.4	170	262	236	210
58	120.1	107.9	96.1	180	279	251	223
59	121	108.7	96.8	190	297	267	237
60	121.8	109.5	97.5	200	315	283	252
61	122.7	110.4	98.2	210	337	303	269
62	123.7	111.3	99	220	360	324	288
63	124.6	112.2	99.7	250	424	381	339
64	125.6	113.1	100.4	300	557	500	445
65	126.5	114	101.2	350	717	655	573
66	127.5	115	102	400	913	821	731
67	128.4	116	102.8	450	1140	1025	911
68	129.3	117	103.5	500	1418	1275	1133
69	130.3	118	104.3	550	1735	1560	1390
70	131.2	119	105	600	2330	2095	1865

Calculation of Heat Requirements

The calculation of the heat requirements is, therefore, quite different in principle from those usual for convective heating. Instead of calculating the heat lost by conduction through the walls, an endeavor is made to estimate the surface temperatures which the various parts will assume. If the surfaces are too cold they will absorb too much radiant heat from the body. By introducing a number of smaller surfaces maintained at a high temperature and disposed in suitable positions about the room, the negative radiation from the cold walls and windows is neutralized, and the mean radiant temperature is raised to the desired value. The problem, therefore, is first to find the value of the radiant temperature which would absorb a desirable amount of radiant heat from the body and then to determine a suitable amount of surface, maintained at a suitable temperature and placed in suitable locations to raise the mean radiant temperature to that desired value.

The next step is to consider the loss of heat from the body. The amount given off as radiation by any body to an environment at absolute zero is given in Table 1 and shown graphically in Fig. 2. The second column of Table 1

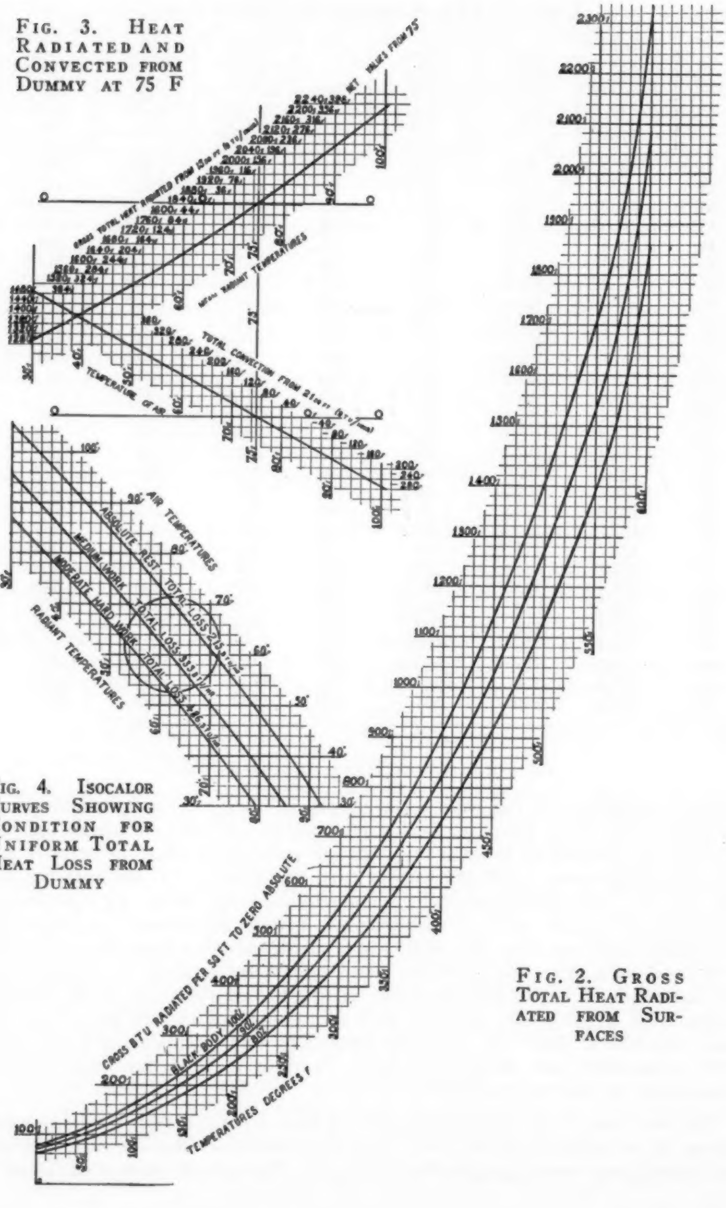


FIG. 3. HEAT RADIATED AND CONVECTED FROM DUMMY AT 75 F

FIG. 4. ISOCALOR CURVES SHOWING CONDITION FOR UNIFORM TOTAL HEAT LOSS FROM DUMMY

FIG. 2. GROSS TOTAL HEAT RADIATED FROM SURFACES

TABLE 2. HEAT LOST PER HOUR BY CONVECTION FROM VERTICAL CYLINDER 12 IN. DIAMETER OF 21 SQ FT AREA MAINTAINED AT 75 F TO SURROUNDING AIR AT VARYING TEMPERATURES

Temp. of Surrounding Air	Diff. from 75 F	Total Heat Lost by Convection Btu per Hour	Temp. of Surrounding Air	Diff. from 75 F	Total Heat Lost by Convection Btu per Hour
30	−45	490	75	+ 0	− 0
35	−40	432	76	+ 1	−11
40	−35	378	78	+ 3	−32
45	−30	324	80	+ 5	−54
50	−25	270	82	+ 7	−75
52	−23	248	84	+ 9	−97
54	−21	227	86	+11	−119
56	−19	205	88	+13	−140
58	−17	183	90	+15	−162
60	−15	162	92	+17	−183
62	−13	140	94	+19	−205
64	−11	119	96	+21	−227
66	− 9	97	98	+23	−248
68	− 7	75	100	+25	−270
70	− 5	54			
72	− 3	33			
74	− 1	11			
75	− 0	0			

shows the amount given off per square foot per hour from an absolute black body and the third the amount given off from a body having a surface radiativity of 90 per cent. The net amount given off or absorbed by any surface when completely enclosed in a chamber whose walls are at a different temperature is the difference between the amount which it itself emits at the higher and the lower temperatures respectively. Thus a body of 90 per cent radiativity at 75 F contained in an enclosure at 63 F would emit a net amount of heat of 123 − 112.2 = 10.8 Btu per square foot per hour.

The human body is of varying size and shape due to its constant movement, and of varying temperature. It is therefore necessary to assume for purposes of calculation that it is represented by some solid object of geometrical form maintained at a constant temperature of which the fixed dimensions are known, and from which the amount of heat lost can be calculated. Complication arises from the fact that the surface for purposes of radiation is not the same as that for purposes of convection.

The whole of the outside surface loses heat by convection but a considerable area is protected from radiation by its re-entrant shape. Thus the areas between the legs, or under the arms, or between the head and the trunk and other parts radiate chiefly to one another and therefore, the amount of radiation from those portions is nullified or greatly reduced.

From calculations it has been estimated that in respect to radiation an average body may be represented by a cylinder having a diameter of 12 in. and an area of 15 sq ft, and for purposes of convection by a similar cylinder having a superficial area of 21 sq ft. (See Table 2.)

Heat Loss

Rubner's figures show that 338 Btu per hour are lost from a normal human body at ordinary sedentary work when maintained in a state of comfort. Of this total the same authority estimates that about 195 Btu are lost by radiation and 138 Btu by convection, presumably for the actual conditions of the room in which the experiments were made of which he gives no particulars.

If a body of 15 sq ft maintained at 75 F is to lose 195 Btu per hour by radiation,

$$\frac{195}{15} = 13 \text{ Btu per square foot per hour,}$$

the environment must have an emission of $123 - 13 = 110$ Btu per sq ft per hour, which corresponds to a mean radiant temperature of 60.5 F (see Table 1).

Similarly, if a cylinder having an area of 21 sq ft is to lose 138 Btu per hour or $\frac{138}{21} = 6.6$ Btu per square foot per hour, this corresponds to an air temperature of 63 F. It therefore appears from these figures that the problem is to raise the radiant temperature to 61 F and the air temperature to 63 F.

Example

The following example of a small room will illustrate the calculations for the radiant temperature. The figures given in Table 3 for the temperatures are

TABLE 3. SURFACE AREAS, TEMPERATURES AND EMISSIONS FOR A ROOM OF 5760 CU FT

	Area Sq Ft	Assumed Surface Temperature (Deg Fahr)	Heat Emission (Btu per Sq Ft per Hr)
External Wall	297	50	101.5
Glass	279	45	97.7
Inner Wall	480	55	105.5
Ceiling	480	55	105.5
Floor	480	55	105.5

appropriate for the English climate. Correspondingly lower figures would have to be determined for colder climatic conditions.

Calculations for mean radiant temperature (MRT):

$$\begin{array}{rcrcr}
297 & \times & 101.5 & = & 30{,}000 \\
279 & \times & 97.7 & = & 27{,}300 \\
480 & \times & 105.5 & = & 50{,}600 \\
480 & \times & 105.5 & = & 50{,}600 \\
480 & \times & 105.5 & = & 50{,}600 \\
\hline
2016 & & & & 209{,}100
\end{array}$$

$$\text{Mean emission} = \frac{209{,}100}{2016} = 103.5 \text{ Btu per square foot}$$

Corresponding MRT = 53 F

If a body of 15 sq ft at a surface temperature of 75 F were put in this room the heat it would lose would be $15 \times (123 - 103.5) = 290$ Btu per hour, which is 95 Btu per hour or 6.35 Btu per square foot per hour too high.

In order to raise the MRT to 60.5 F proceed as follows:

Assume the surface temperature of the hot plate to be 200 F which corresponds to that of a steam heated metal plate as used with this type of heating system and which will be discussed later. The summation representing the room calculations instead of being 209,100 Btu should be $2016 \times 110 = 221{,}760$

Fig. 5. London General Omnibus Factory Showing Metal Plates on Roof

Btu so it is necessary to increase the 209,100 Btu by 12,660 Btu. The emission per square foot at 200 F is seen to be 283 Btu. The required surface is therefore $\frac{12{,}660}{283} = 45$ sq. ft. If the surfaces are 30 in. wide the total length required is about 18 ft. The effect of this surface suitably disposed would be to raise immediately the radiant temperature to the required degree and to maintain it at that value as long as the surfaces remained at the values assumed.

It is necessary also to calculate how much heat will be given off by the same surfaces by convection and thereby to ascertain whether this amount of convected heat is adequate for warming the air corresponding to the degree of

ventilation required. If it is not, then additional convection surfaces must be introduced to make up the balance.

One peculiarity of radiant heat is that it is practically used twice, once in maintaining the radiant temperature and once by what may be called secondary convection. Any heat falling on the furniture, carpets, or walls, thereby raises the temperature of their surfaces, and that heat is partly used in raising the temperature of the air which comes in contact with them. It is not possible to

FIG. 6. ENTRANCE HALL IN PRIVATE HOUSE SHOWING METAL PLATES USED ORNAMENTALLY

make any rational calculations of the amount of heat so communicated to the air, which depends among other things on the character and disposition of the furniture and on the conductivity of the walls. It appears that this factor can only be determined empirically and a good deal of observation and research is necessary in the climatic conditions concerned before any authoritative figures could be established.

Such is, briefly, the principle on which these calculations are made. It will be seen that it is possible to prepare a table which will give the amount of

heating surface at high temperature which will compensate for a square foot of cold surface at a lower temperature, so that the calculations are very much simplified. Once the quantities are calculated in this way the ingenuity of the engineer must be exercised in order to find means for supporting the necessary panels in suitable positions.

Practical Applications

The manner in which these principles may be carried out in practice offers a wide scope for ingenuity and invention. There are three main features of any method:

1. The character of the radiating surface which is to be kept hot.
2. The method of supporting it and its disposition.
3. The means of conveying heat to it.

In England the author has been responsible for most of these proposals. The first one consists of embedding coils of pipes in the concrete of the ceilings or the surface of the plaster and forcing hot water or in some cases steam through the coils. This has the effect of heating the whole of the plaster or concrete in contact with the pipes. This method has been successfully applied to at least two American buildings one of which is the British Embassy at Washington.

Sundry improvements have been made in England to the original method with the object of preventing or restricting the transmission of heat into the body of the concrete, and so raising the temperature of the plaster surface. This has been done in some cases by forming recesses in the concrete which are filled with insulating material, on which the pipes are laid, being later covered in by the plaster in various ways so that the pipes are not immovably embedded in the concrete.

Iron Plates

To accomplish the same purpose, metal plates have been extensively used in several different ways. The most favored method is by forming a radiator with a flat front face with shallow waterways cast on them at the back in which hot water or steam circulates or in which electric elements are fixed. These plates are made in sections and are nippled together just as is an ordinary radiator, leaving the front face flat. These surfaces are either attached direct to the face of the wall or ceiling, or in many cases installed in specially formed recesses, so as to present a uniform finished surface. There are several thousands of such installations in England, photographs of a few of which are shown in Figs. 5 to 10.

In some cases in order to increase the convection, and thereby the total heat output, a space is formed at the back of the metal plate so that air can circulate over both the front and the back face. Though this has the effect of reducing the cost of the plant and the necessary surface, it increases substantially the fuel consumption. Anything which tends to increase the amount of convection tends pro rata to increase the fuel consumption and to decrease the comfort. In some cases the flat surface of the metal plate has been covered with wall-paper or other decorative materials.

Veneers

Removable wood or composition panels are sometimes placed over the flat plate and these become warm by contact, special arrangements being made to prevent warping and other undesirable effects.

Floor Heating

A similar principle has also been used in England to a considerable extent in which all the heat is communicated to the floor. The most notable example of this is the new Liverpool Cathedral which is entirely heated in this way. It is also done in smaller buildings such as schools and churches, by covering

FIG. 7. HOSPITAL WARD SHOWING METAL PLATES AS DADO

the floor with removable precast slabs supported on dwarf walls, thus producing a series of channels under the floor. In these channels pipes, heated either by water or by electricity, are installed and, being insulated on the under side, communicate a large part of the heat given off to the slabs immediately over. These are raised to about 70 F and the effect is very pleasing; a feeling of warmth and comfort is produced even though the surrounding air is quite cold.

Heating Media

The media by which the heat is conveyed to the plate are as follows:

1. *Steam,* which is rarely used in England except in workshops and factories.
2. *Hot water,* which is circulated to the surfaces by centrifugal pumps.
3. *Electricity,* which is now extensively used in England for heating purposes.

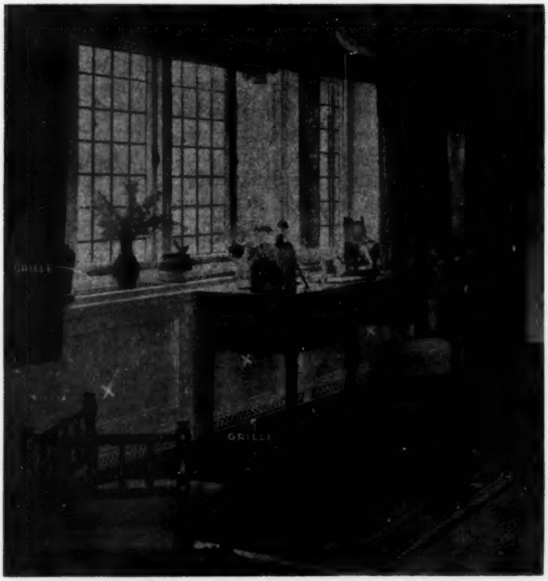


FIG. 8. SHOWROOM ILLUSTRATING METAL PLATES COVERED WITH VENEER

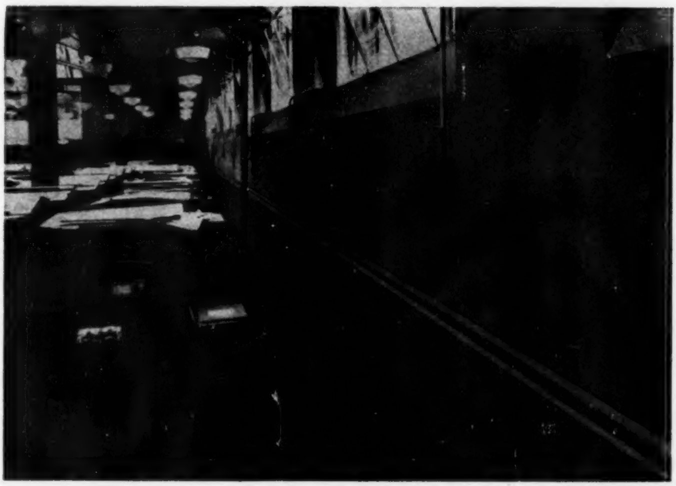


FIG. 9. DRAWING OFFICE OF AMERICAN FIRM IN LONDON HEATED WITH METAL PLATES

There are more than 3,000 installations in England heated by electricity. This is done in several different ways. The most successful up to the present has been to use a specially-designed metal plate of the same general construction as has been described, with a specially-designed flexible electrical element in each of the passages. The heat from these elements rapidly communicates itself to the iron and spreads by conduction over the front surface. Such plates are usually applied to the ceiling as shown in Fig. 11, more rarely to the walls. In some cases they are installed as a continuous dado along the whole length of a room, as for instance, in a hospital ward. (See Fig. 7).

Fig. 9 shows a similar arrangement in which a dado is formed along a drawing office. This particular case is the property of an American firm

FIG. 10. HOSPITAL WARD WITH METAL PLATES ON CEILING

which has established a branch in England. Their initial requirements called for a maintained temperature of 75 F. When it was explained to the clients that in England such a temperature would be altogether excessive and intolerable, they consented, under protest, to reducing the demands to 70 F, to which value the calculations were made. When the apparatus was first put into use two years ago it was found that any air temperature approaching 70 F was far too high and instructions were given that it be reduced by maintaining a temperature of the water leaving the boiler of 120 F, at which temperature the interior of the room is maintained in cold weather at about 65 F, which is found to be as high as can be endured.

Another method which has attained wide use in England is a pottery panel in which an electrical heating element is incorporated, the whole being cast in

one solid block. The heating element consists of a graphite grid connected to two conductors on the outside of the panel. These can be raised to a fairly high temperature, about 500 F, and are located in the corners and on the walls of a room from which they radiate their heat into the room interior. This is a very inexpensive method of carrying out this principle. Fig. 12 shows the interior of a school room in England so heated.

All these panels are controlled by thermostats such that when the radiant temperature in the room reaches a pre-determined value the panels are automatically switched off one at a time so that the temperature is maintained uniform, and no more electrical power is used than is necessary for the purpose. It is found that a building can be heated in accordance with the requirements of

FIG. 11. RANGE OF OFFICES SHOWING ELECTRICALLY HEATED METAL PLATES ON CEILING

the English climate throughout an entire winter season, for an expenditure of electrical energy equal to about 1 kwhr per cubic foot of space heated, and for less than half this expenditure for buildings such as schools or churches which are not continuously in use.

This method has useful application in churches and in similar buildings in which the conditions can be localized, so that the people sitting in the seats are kept warm without warming the rest of the building. In these cases electrically heated tubes are installed under the seats, and in some cases pottery panels are fixed in such positions as to radiate downwards on the bodies of the congregation.

Comfortable conditions can thus be established even on a very cold day by switching on the electrical power about 2 hr before the congregation is due to assemble. It is found that even with the high cost per Btu of electrical power, as compared with coke, the total annual cost of heating a church in this manner is considerably less than heating by coke. Electrical energy is supplied for the

purpose at 1¢ per kilowatt hour. The most desirable method of heating a building by electrical power involves a combination of all these three methods, and much experience has accumulated in England on this subject.

In England there is a greatly increased use of this method in what are called open air schools, which are schools with wide opening windows on each side, and in all ordinary weather these are kept wide open, so that there is a very free passage of air through the class rooms. In these conditions

FIG. 12. SCHOOLROOM HEATED WITH ELECTRICAL CLAY PANELS

the heat can only be delivered by radiation, as all convected heat would at once be blown away.

Gas

The author has also made attempts to carry out the same principle by heating radiating panels locally by a small gas jet. The products of combustion in this case are drawn through the passage and off the outlet by means of a small centrifugal fan, located in the basement, or in any other suitable position. A system of copper pipes is carried from the outlet to the fan and the diameters are so calculated as to maintain a suction of about ½ in. of water at the outlet of each of the metal plates. The effect is very good and the economy in gas is considerable. It is found that not more than half as much gas is required for heating a building in this way as by a system of gas-heated radiators in which the products of combustion are allowed to escape into the room. The latter system has some vogue in Great Britain, though it is not one which some of us would recommend on account of the undesirable introduction into the air of the room of the products of combustion and much water vapor, but in some situations it is found reasonably satisfactory in England.

DISCUSSION

L. J. FOWLER (WRITTEN): In the paper of Arthur H. Barker he tabulates five formidable risks which are likely to arise where "pipes are embodied permanently and immovably in concrete."

We would point out that in the whole of the installations which have been carried out on this system, which comprise about 7½ million feet run of panel piping embodied in concrete, no defect has arisen from any of the risks referred to. A point we desire to emphasize is that one of the essential features of panel heating is the low surface temperature at which the system works. Any surface temperature above 100 F is liable to cause discomfort, and for this reason the metal panels worked at a high temperature cannot be considered as in any way representing the principles known as panel heating.

We are the "very enterprising and able firm of heating engineers" referred to, and while it is true that we were entirely responsible for the development of panel heating using, among others, one of Mr. Barker's patents, it is untrue that we took any "terrifying risks."

The very first installation of panel heating, the Royal Liver Building, Pier Head, Liverpool, made in 1908, must be known by sight to thousands of American citizens; this was a most successful installation from its inception and is still working, giving complete satisfaction.

PRESIDENT CARRIER: The system of heating just described is practically unknown to people of America. It has been developed in England by one of our members who has written this paper, Room Warming by Radiation. He has come from England to present it: A. H. Barker, consulting engineer in London for many years and a Past President of the *Institution of Heating and Ventilating Engineers* officially representing that organization. It was a great pleasure to hear Mr. Barker and I compliment him for his very able exposition of a little known subject. I would like to point out to our members that this method of heating is, I believe, ideal for certain climatic conditions from the standpoint of economy and also comfort. I think that will be particularly true of climates like England and probably all of the southern half of our United States.

There is, however, a definite relationship between the economy of heating and maximum temperature difference, and also a definite relation between economy and the room temperature maintained with reference to the outside. The lower the effective temperature of the room, including radiation and convection, and the higher the outside temperature, the greater is the economy with this system. One reason is that with fairly good ventilation and low temperature differences between the outside and indoor air, ventilation losses are almost eliminated by the radiation method.

The loss in heating up air is a very important factor. To get ventilation it is unnecessary to heat up that air. Mr. Barker referred to this, and it is, in my opinion, the place where the great economy is effected. As the temperature drops the gain in percentage in loss of ventilation decreases, so that at low outside temperature there is not the same proportionate economy with this system, I believe, as with others. That is just mathematics.

There is one other point that I wanted to demonstrate on the blackboard. This has nothing to do with the type of installation, but is simply the principle I have tried to get over a few times, but with indifferent success. That is the question of thermometry which underlies this whole discussion. I think we want to get on firm ground on that and not think too vaguely in the matter. I have wanted to discuss this paper first for that reason.

There is of the total sensible heat removed from the human body a little less than

60 per cent removed by radiation and a little more than 40 per cent by convection. There is also from 12 per cent to 25 per cent lost by evaporation. The latter factor depends upon the air temperature and the wall temperature, and the figures given are for 70 deg or lower. If we consider the total loss from the body under conditions of comfort, we will have approximately 45 per cent lost by radiation, 30 per cent lost by convection, and 25 per cent lost by evaporation.

Let us determine now the relative effect of radiation and convection, first upon the thermometer and second upon the heat removal from the human body. We know that in a black body we have a division between radiation and convection when the air and walls are of the same temperature in the proportion of about 60 to 40 as above. At low temperatures a thermometer acts substantially as a black body, as also does the Kata-thermometer developed by Dr. Leonard Hill. However, when the temperature of the wall surfaces and the air of the room are different, then the proportions of radiation and convection losses are obviously modified. For example, if the temperature difference between the body and the walls was 20 deg, and between the body and the air was 30 deg, we would have:

$$\begin{aligned} 25 \times 0.60 &= 15 \\ 37\tfrac{1}{2} \times 0.40 &= 15 \\ &\overline{30} \end{aligned}$$

which would show that the radiation effect would then be exactly equal to the effect of convection at the lower air temperature. It would also show that the effect would be identical with that if the air and wall surfaces were both at the same temperature of 24 deg below the body temperature as for example:

$$\begin{aligned} 30 \times 0.60 &= 18 \\ 30 \times 0.40 &= 12 \\ &\overline{30} \end{aligned}$$

Let us now apply these effects to a thermometer instead of a human body or a Kata-thermometer. Let us assume that the thermometer acts as a black body, and also assume that the radiation effects are directly proportionate to the temperature difference, which is approximately true for small temperature differences. It is obvious that the thermometer being a neutral body giving off no heat of its own must receive by radiation just as much heat as it loses by convection, or vice versa, where the surrounding wall temperatures and the air temperature in the enclosure are different.

Taking the illustration in which the wall temperature is 12½ deg above the air temperature, we will have this difference divided in the ratio of 40 per cent and 60 per cent. In other words, the thermometer will be 5 deg lower than the wall temperature and 7½ deg higher than the air temperature. As a check:

$$\begin{aligned} -5 \times 0.60 &= -3 \\ +7\tfrac{1}{2} \times 0.40 &= +3 \\ &\overline{0} \end{aligned}$$

That is, the heat flowing to the thermometer by radiation exactly equals the heat lost from the thermometer to the air from convection. Now, if we assume the wall temperature to be 73 deg and the air temperature to be 12½ deg lower, or 60½ deg, then the thermometer would read 68 deg which is 5 deg lower than the wall temperature and 7½ deg higher than the air temperature.

It is also seen from the above that, assuming the body surface temperature to be 98 deg, we would have the same sensible heat loss and, therefore, the same sensation

of comfort provided the wet bulb temperature was the same in both cases whether the air and the walls were both 68 deg, or whether the walls were at a temperature of 73 deg and the air at a temperature of 60½ deg. Also, the dry bulb thermometer would read the same in both cases if stationary. This relationship, of course, is only true for still air, as moving air would increase the proportion of convection to radiation, but it should be pointed out that it should do so equally so far as the sensible heat effect is concerned both for the human body and for the thermometer.

Taking now a more extreme case and assuming our wall temperature was 78 deg, or 20 deg below the assumed body temperature, and the air temperature was 53 deg, or 35 deg below the body temperature, then, as before, our thermometer would read 68 deg, or 10 deg below the walls and 15 deg above the air temperature. The same calculations will show that the total of radiation and convection will be approximately the same as though both room and air were at 68 deg.

Inasmuch as it is a fact that the sensible heat loss is dependent upon the difference between the body temperature and the dry bulb thermometer as affected both by radiation and convection, and the tendency to evaporation is dependent *only upon the wet bulb temperature,* then it is true, providing both wet and dry bulb thermometers read the same at a given location in the room, that it matters not what the respective temperatures of the air and the surrounding walls are. Any change due to raising the surrounding wall temperature can be compensated for by a lowering of the air temperature; and any combination of wall temperature, air temperature, and wet bulb temperature can be made to read direct upon the present comfort chart simply by observing the wet bulb temperature and dry bulb temperature as before with still thermometers.

A. J. Offner: The paper presented by Mr. Barker is timely. While it lacks something in the physics and theory of radiation, it clearly shows the calculations of and the applications of radiant heat to room warming. It is hoped that the paper will start constructive thought and discussion as to what is really the proper method of applying the heat to the space to be warmed, not only as to personal comfort but also as to health, temperature and condition of the air, cleanliness and economy.

The purpose of the first room heater, whether in the form of horizontal pipe coils or vertical pipes screwed into a cast iron base, apparently was to heat the room air to 70 deg, and when that temperature was reached and maintained the room was considered properly heated. In that classic of early heating textbooks, Steam Heating for Buildings, written by the late Wm. J. Baldwin, and I believe first published in 1881, radiators are described as those *heaters placed within a room or building to warm the air.* In this same book, in describing the difference between direct and indirect radiation, direct radiators are stated to be those which *warm only the air in the room and maintain the heat.* While the pioneers of our profession apparently thought of room heater performance only in terms of air temperatures, they insisted on calling them *radiators.* Even to this day, there are still many who only think of radiator performance in terms of air temperatures.

Webster defines a radiator as *that which radiates or emits rays, especially that part of a heating apparatus designed to radiate heat,* while convection is defined as a *process of transmission as of heat or electricity by currents in fluids.* Therefore a room heater which does nearly all of the heating by radiation may correctly be called a radiator, while those heating and circulating the air are really convectors. Of course, every form of room heater does heating by both radiation and convection, but in various proportions depending on type and application.

From the research laboratory of our Society and the experimental work of others, it is well known that air temperature is not the only criterion of comfortable and healthful heating, but that there are other factors. It is also well known that it is possible to stand unclothed in bright sunlight, with snow on the ground and very low air

temperature, and still feel comfortable providing, of course, there is no wind or air movement.

Recent developments in types of room heaters have not only improved the old style radiator, both as to appearance and efficiency, but also have introduced heating units transmitting the heat either by radiation or by convection, or a combination of the two in various proportions. The system of heating by large heated surfaces at low temperatures approaches nearest to true radiators, while those systems using warm air heated either by natural circulation through enclosed heaters or by mechanical means are convectors. The sectional cast-iron radiator transmits heat by both radiation and convection. A recent development of this sectional type, having a flat front, increases the amount of radiant heat given off by this kind of radiator. Should future development of room heater design tend more towards radiant heat, with corresponding low air temperatures or towards convected heat with higher air temperatures, with or without humidity control? Personally, I am inclined to lean towards a combination of more radiation, less convection, lower air room temperatures and humidity control. Further research along these lines should be of great value and interest.

I would like to add one point to the above, which might be of interest. About 25 years ago two heating systems were designed, one by Konrad Meier, who is a member of this Society now living in Switzerland, in which the principle of panel heating was applied: one of the buildings being the Lying-In Hospital in New York City; the other the Psychiatric Clinic building at the Johns Hopkins Hospital in Baltimore. In certain special rooms of these two buildings the heating units consisted of steel plates carried around the outside walls of the building, back of which were located a series of pipes containing hot water. Therefore, even here in the United States, room warming by radiation using panels is not new.

DR. C. W. BRABBÉE: If I may refer to Mr. Barker's first remark, I feel myself transferred to Buffalo, where in 1925 I discussed the very same subject, only he stated it very much better. When I see some of the pictures I feel myself moved to my office in New York where we have exactly the same panel heat under the windows and everybody who has seen this installation has agreed it is the finest method of heating.

However, when I came from abroad I was told that America wants what she wants and she gets what she wants, right or wrong. In the building industry the architect is king and the consulting engineer viceroy. The architect says, "I give you very little space for your heating today. Tomorrow I give you nothing." The consulting engineer says, "We have tremendous temperature differences in this country, a 40 deg drop and 60 deg below zero sometimes. I must heat my building so there will be no complaints. I must put in plenty of heating surface."

Now plenty of heating surface and no space are things which are detrimental to the kind of heating described by Mr. Barker. If he can get the architect's support, then those difficulties might be overcome.

L. A. HARDING: May I be permitted to make a motion? The motion reads as follows:

The Secretary be instructed to extend to Mr. Barker the appreciation of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS for the presentation of his most excellent and instructive paper.

The motion was seconded and carried.

No. 927

PERFORMANCE OF CONVECTOR HEATERS

By A. P. Kratz [1] (*MEMBER*) and M. K. Fahnestock [2] (*NON-MEMBER*)
Urbana, Ill.

This paper is the result of research conducted at the University of Illinois in cooperation with the A. S. H. V. E. Research Laboratory

The data presented in this paper were obtained in connection with an investigation conducted by the Engineering Experiment Station of the University of Illinois, of which Dean M. S. Ketchum is the Director. This work is carried on in the Department of Mechanical Engineering under the general supervision of A. C. Willard, Professor of Heating and Ventilation and head of the Department. Acknowledgment is due E. L. Broderick, Research Assistant in Mechanical Engineering for valuable assistance in the collection and preparation of the data.

The object of this part of the investigation was to make a comparison of the performance of four types of cabinet convector heaters with the performance of an unenclosed direct cast-iron radiator of approximately the same height when a temperature of 68 F was maintained at a level 30 in. from the floor in all cases. A further object was to compare the performance of the convector heaters with that of a tubular cast-iron radiator enclosed in a cabinet conforming to the space requirements of the convector heaters, and operated under the same conditions.

Description of Apparatus

All tests were conducted in the low temperature testing plant which has been completely described in previous publications.[3] In brief, this plant, as shown in Figs. 1 and 2, consists of two rooms erected within a larger room having

[1] Research Professor in Mechanical Engineering, University of Illinois.
[2] Special Research Assistant Professor in Mechanical Engineering, University of Illinois.
[3] University of Illinois Engineering Experiment Station Bulletins Nos. 192 and 223. Also Transactions of the American Society of Heating and Ventilating Engineers, Vol. 35, 1929, pages 79-89.

Presented at the 38th Annual Meeting of the American Society of Heating and Ventilating Engineers, Cleveland, Ohio, January, 1932, by A. P. Kratz.

6-in. cork walls, floor and ceiling. The latter room is equipped with direct expansion refrigerating coils, and each of the two test rooms presents two walls of standard frame construction exposed to any desired temperature maintained in the cold room. One wall of each test room contains a pair of double

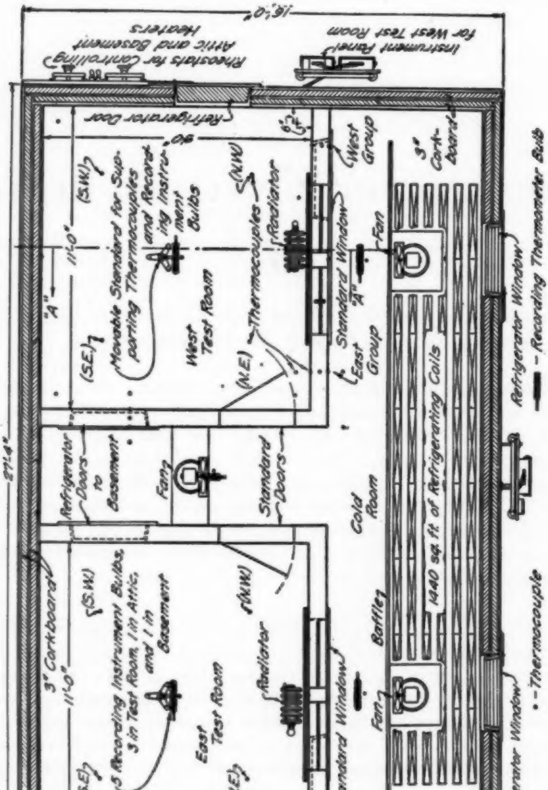


FIG. 1. PLAN SECTION OF LOW TEMPERATURE TESTING PLANT

hung windows and the other wall a standard door with a glass panel. In addition, arrangements are made so that the floors and ceilings of the test rooms may be exposed to any desired temperatures. Air movement over the walls is obtained by means of fans.

The plant is thoroughly equipped with thermocouples and temperature recorders in order to permit observations being made without the necessity for entering the rooms and thus disturbing conditions at any time during the period of a test. The radiators are connected with a single pipe for steam and return

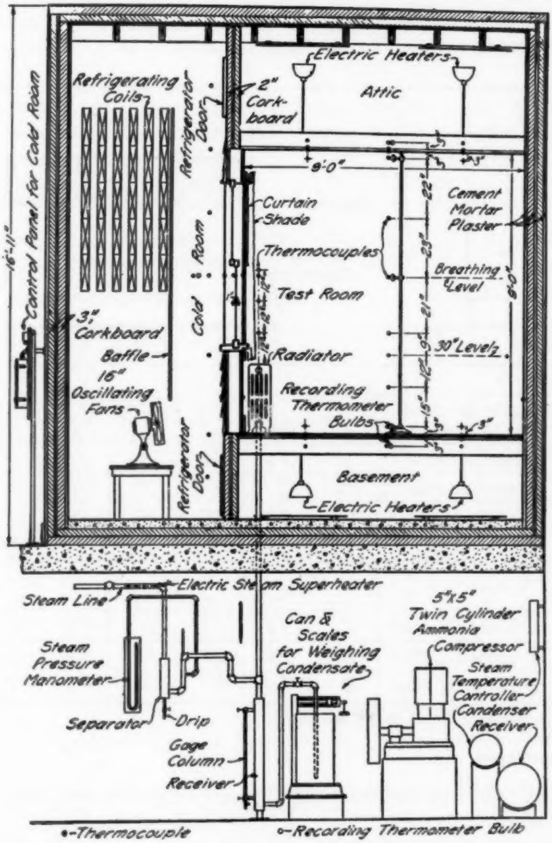


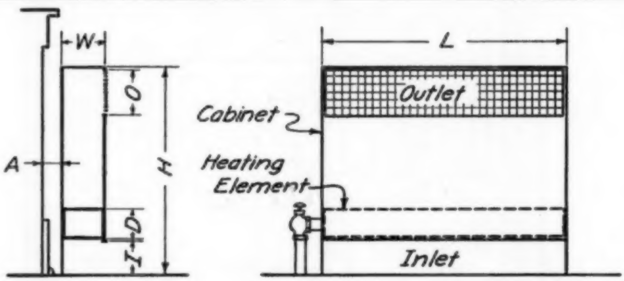
FIG. 2. ELEVATION SECTION OF LOW TEMPERATURE TESTING PLANT

condensation, and the condensation is weighed as shown in Fig. 2. A steam separator is installed in the line to each radiator, and an electrical superheater further insures the use of dry steam. Provision is also made for venting the radiators to prevent the accumulation of air.

The temperatures at the various levels within the test rooms are observed by means of thermocouples made from No. 22 B and S gage copper and con-

stantan wire and supported on a standard in the center of the room as shown in Fig. 2. A study of temperature conditions within the test rooms proved that the center reading at a given level is representative of the average temperature at that level.

Four representative convector heaters of the cabinet type were used. The dimensions of the cabinets are shown in Fig. 3, and further details of the heating elements are shown in the insets in Figs. 4 to 7. Convector heater



Convector Heater No.	Type of Heating Element	Dimensions in Inches						
		A	W	H	L	I	O	D
1	Cast Iron	$2\frac{1}{2}$	$5\frac{1}{2}$	26	36[1]	4	5	$14\frac{3}{16}$
2	Non-ferrous	$2\frac{1}{2}$	5	$30\frac{3}{8}$	31	6	6	6
6	Non-ferrous	$2\frac{1}{2}$	$5\frac{1}{2}$	28	30[2]	4	5	2
7	Bimetallic	$2\frac{1}{2}$	$6\frac{1}{16}$	$28\frac{3}{8}$	$22\frac{5}{8}$[2]	$4\frac{1}{8}$	$3\frac{1}{8}$	20
Enclosure No.								
24	3-Tube C.I. Rad.	2	$5\frac{3}{4}$	27	$31\frac{1}{2}$	$4\frac{1}{4}$	6[3]	26
24d	3-Tube C.I. Rad.	2	$5\frac{3}{4}$	27	34	$4\frac{1}{4}$	6	26
24e	3-Tube C.I. Rad.	2	$5\frac{3}{4}$	27	34[4]	$4\frac{1}{4}$	$22\frac{1}{4}$	26

[1] 6" of length blocked. [2] 3" of length blocked.
[3] Open outlet.
[4] 1-Section Wrapped.

2-10-32

FIG. 3. DIMENSIONS OF CONVECTOR HEATERS AND ENCLOSED CAST-IRON RADIATOR TESTED

No. 1 contained a cast-iron heating element with cast-iron fins. Heaters Nos. 2 and 6 contained non-ferrous heating elements with thin metal fins. Heater No. 7 contained a non-ferrous heating element with thin metal fins, combined with a cast-iron front, the latter partly filled with steam and serving as direct panel radiation.

In addition, tests were run on a 13-section, 26-in., 3-tube cast-iron direct steam radiator enclosed in a cabinet similar in design and space requirements to those used in connection with the convector heaters. Three types of cabinets were used with this radiator, as shown in Fig. 3, and in the inset in Fig. 8. They

consisted of (1) Enclosure No. 24, a cabinet with open inlet and outlet, (2) Enclosure No. 24d, with an open inlet and grilled outlet, and (3) Enclosure No. 24e, with a fully grilled front extending to within 4¼ in. of the floor. An 8-section, 26-in., 5-tube unenclosed direct cast-iron radiator was also tested, and was used as a standard for comparison in all cases.

TEST PROCEDURE

In all cases, the temperature in the cold room was maintained at about —2.0 F and one of the exposed walls of the test room was subjected to an equivalent wind velocity of approximately 10 mph. The temperature of the air

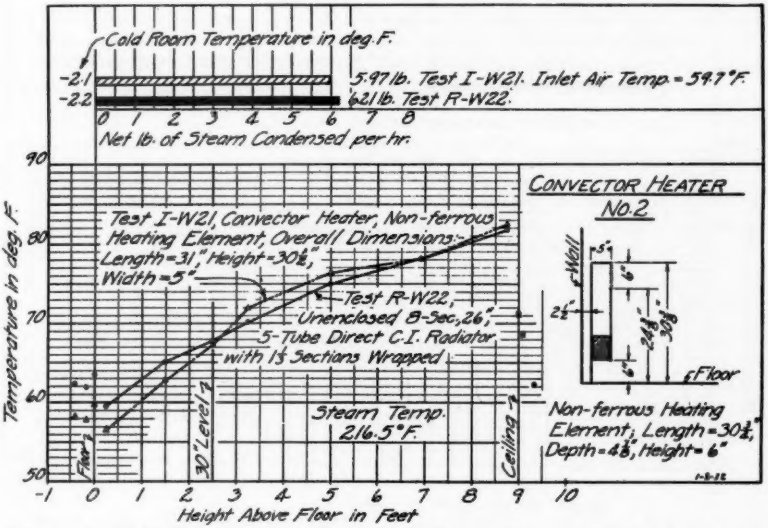


FIG. 4. ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR AN UNENCLOSED DIRECT C. I. RADIATOR AND A CONVECTOR HEATER WITH A NON-FERROUS HEATING ELEMENT WITH THE SAME TEMPERATURE AT THE 30-IN. LEVEL

above the ceiling of the test room was maintained at 62 F and the air in the basement at such a temperature that the upper surface of the floor was approximately two degrees warmer than the lower surface. No test observations were made until conditions had remained constant for several hours, as indicated by the readings of the thermocouples on the inside wall surfaces. This required about 20 hours after a change in radiators was made. When the required thermal constancy had been attained, the condensate was weighed over a period of one hour, and no test was accepted if the condensate showed more than 2½ per cent deviation in the successive 10-min. increments of weight. At the end of each test a separate test was run to determine the condensation in the piping alone, and the total condensate was corrected by subtracting the amount so determined.

The output of the heating unit was adjusted, by wrapping part of the sections of the cast-iron radiators or by blocking off part of the cabinets and heating elements of the convector heaters, so that each unit maintained a temperature of 68 F at a level 30 in. above the floor under the standard conditions of exposure previously outlined.

RESULTS OF TESTS

The results of the tests on the convector heaters are shown in Figs. 4 to 7 inclusive. The full line curve in every case represents the performance of the

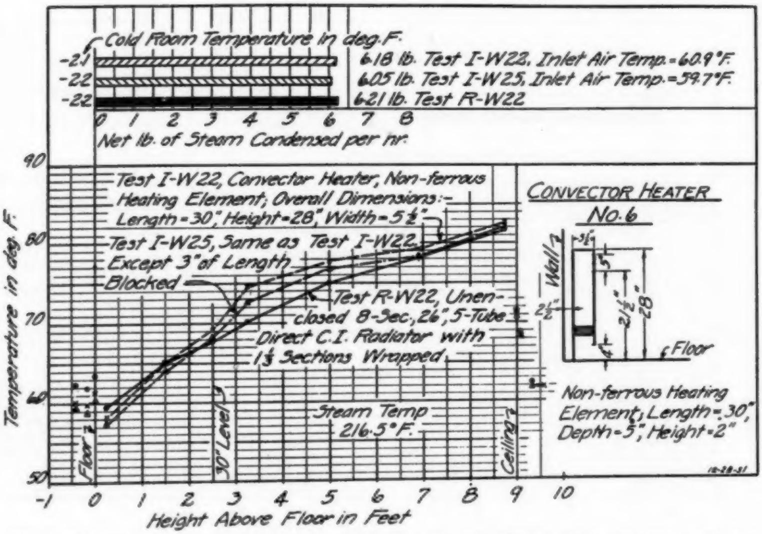


FIG. 5. ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR AN UNENCLOSED DIRECT C. I. RADIATOR AND A CONVECTOR HEATER WITH A NON-FERROUS HEATING ELEMENT WITH THE SAME TEMPERATURE AT THE 30-IN. LEVEL

unenclosed 8-section, 26-in., 5-tube cast-iron radiator with 1⅓ sections wrapped, which was used as the standard for comparison. In Figs. 5, 6, and 7 the temperature gradient curves for convector heaters Nos. 6, 7, and 1 respectively without any blocking are shown. These heaters were chosen from the estimated heating requirements and the stock sizes furnished by the manufacturers. In each case the nearest stock size slightly over-heated the room, and it was necessary to block off part of the cabinet and heating element in order to obtain a temperature of 68 F at the 30-inch level in the test room. The fact that the temperature gradient curves for the blocked heaters were practically parallel to those for the unblocked heaters is evidence that the small amount of blocking necessary had no influence on the characteristic performance insofar as the heating effect was concerned. The comparisons have, therefore, been based

on the steam condensation and temperature gradient curves for the blocked heaters.

From Figs. 4 to 6 inclusive, it may be noted that convector heaters Nos. 2, 6, and 7 showed practically the same characteristics. The convector heater in each case gave somewhat lower temperatures than the unenclosed cast-iron radiator in the zone from the floor to the 30-in. level, higher temperatures in the zone between the 30-in. level and the 7-ft level, and either the same or slightly higher temperatures in the zone above the 7-ft level.

Convector heaters Nos. 2, 6, and 7 maintained a temperature of 68 F at

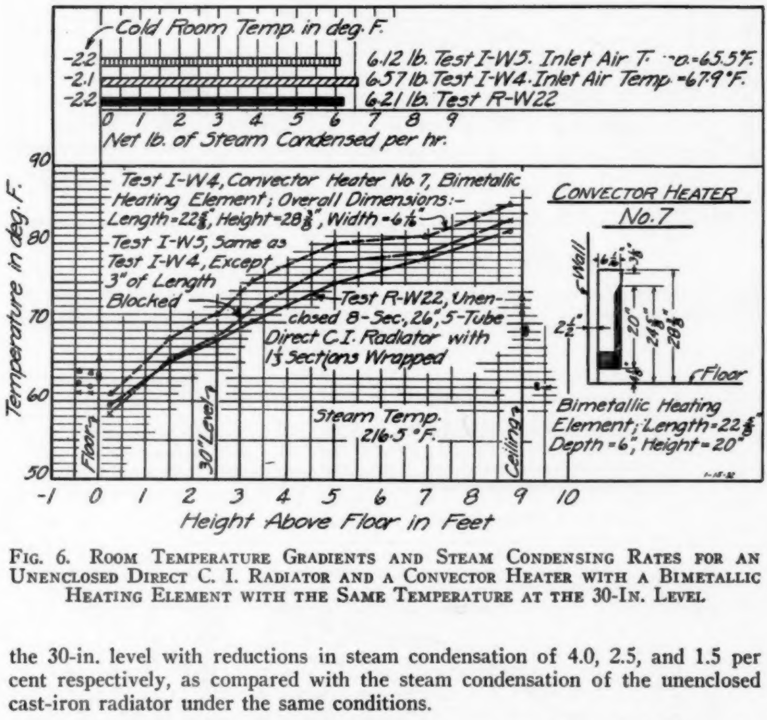


FIG. 6. ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR AN UNENCLOSED DIRECT C. I. RADIATOR AND A CONVECTOR HEATER WITH A BIMETALLIC HEATING ELEMENT WITH THE SAME TEMPERATURE AT THE 30-IN. LEVEL

the 30-in. level with reductions in steam condensation of 4.0, 2.5, and 1.5 per cent respectively, as compared with the steam condensation of the unenclosed cast-iron radiator under the same conditions.

Convector heater No. 1 exhibited the same general characteristics as the others except that the zone of higher temperature extended only as high as the 4.5-ft level and that the temperatures in the zone above this level were lower than those given by the unenclosed cast-iron radiator. Hence, while the resulting comfort conditions were not essentially different from those produced by the other convector heaters, a slight gain in economy was effected by the reduction in the temperatures above the 4.5-ft level. In this case, the reduction in steam condensation was 7.1 per cent as compared with that for the unenclosed cast-iron radiator.

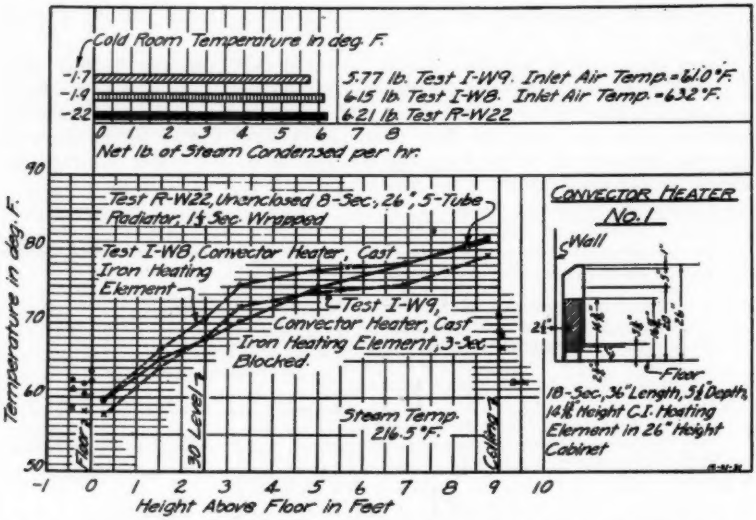


FIG. 7. ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR AN UNENCLOSED DIRECT C. I. RADIATOR AND A CONVECTOR HEATER WITH A C. I. HEATING ELEMENT WITH THE SAME AIR TEMPERATURE AT THE 30-IN. LEVEL

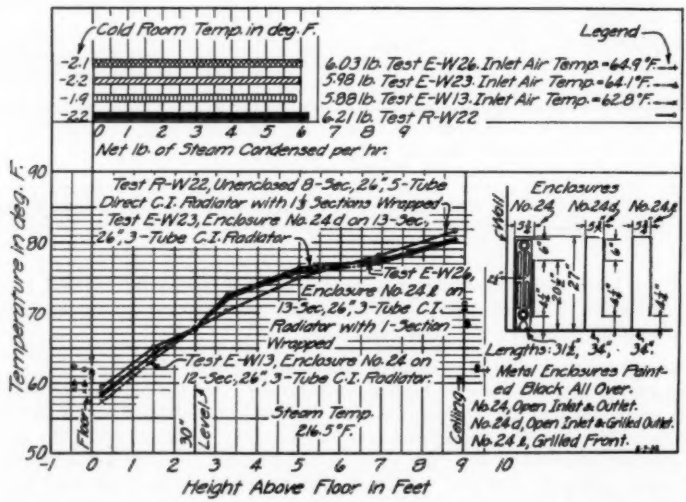


FIG. 8. ROOM TEMPERATURE GRADIENTS AND STEAM CONDENSING RATES FOR AN UNENCLOSED 5-TUBE DIRECT C. I. RADIATOR AND AN ENCLOSED 3-TUBE DIRECT C. I. RADIATOR

The performance curves for the 26-in., 3-tube, cast-iron radiator enclosed in 3 types of cabinets are shown in Fig. 8. These 3 cabinets differed only in the character of the air inlets and outlets, and the over-all dimensions corresponded approximately to those for the cabinets of the convector heaters. It may be noted from Fig. 8 that the performance of the enclosed radiator is compared directly with the performance of the same unenclosed 26-in., 5-tube, cast-iron radiator used in comparison with the convector heaters, and that the enclosed cast-iron radiator exhibited practically the same characteristics as the convector heaters. While some slight adjustments in the size of the radiator with the different types of air inlets and outlets were required, the temperature gradient curves and steam condensations were all practically the same when 68 F was maintained at the 30-in. level. The steam condensation, as compared with that for the unenclosed radiator, was reduced 5.5, 3.8, and 3.0 per cent by Enclosures Nos. 24, 24d, and 24e, respectively. While Enclosure No. 24, with the open inlet and outlet, gave slightly less steam condensation than those with grilled outlets, it also produced somewhat lower temperatures at the floor. On the whole, the performance of the enclosed cast-iron radiator was not essentially different from that of the convector heaters.

Conclusions

From the results of these tests, the following conclusions may be drawn:

1. The 4 types of convector heaters, having cabinets of approximately the same height, did not differ essentially in performance characteristics.

2. The convector heaters produced temperatures slightly lower in the zone from the floor to the 30-in. level, higher in the zone from the 30-in. level to the 7-ft level, and either the same or slightly lower in the zone above the 7-ft level, than those obtained with the unenclosed 26-in., 5-tube cast-iron radiator.

3. The convector heaters maintained a temperature of 68 F at the 30-in. level with steam condensations ranging from 1.5 to 7.1 per cent less than the steam condensation required by the unenclosed 26-in., 5-tube cast-iron radiator.

4. A 26-in., 3-tube, cast-iron radiator enclosed in a cabinet of similar type and conforming approximately with the space requirements of the cabinets used with the convector heaters, exhibited the same performance characteristics as the convector heaters.

DISCUSSION

President Carrier: Is there discussion on this very interesting paper?

Prof. L. E. Seeley: I would like to ask Professor Kratz two questions. Did you put a sheet of metal in back of the bare radiator and test it in that way as compared to the test set-up that you had? In arbitrarily setting that 30-in. height at 68 F, did you have to change the cooling rate of the room and thus probably establish different wall temperatures?

Prof. A. P. Kratz: In answer to the first question, we have placed a metal sheet behind the bare radiator. This procedure effected a reduction in the steam conden-

sation required to maintain the room temperature. In this particular case we were interested only in comparing with the usual setting of the bare radiator and did not place any such sheet.

In reference to our method of running tests, we keep the heat loss from the room practically the same. Stated differently, we adjust the size of the radiator to take care of the heat loss from the room when outside conditions are always maintained constant. That is, zero degrees maintained outside of the exposed walls, 62 F above the ceiling and practically no loss through the floor. The adjustment is made in the size of the radiator to get a balance of heat so that the 68 F is maintained at the 30-in. level.

A. H. BARKER: I would like to ask Professor Kratz, if I may, why he does not adopt the method of measuring the total power electrically. It seems to me that the method of generating the heat by this means is so simple that it facilitates immediate and accurate calculation of the power. It isn't messy like the steam apparatus; you get results straight away. Why don't you apply this method?

PROFESSOR KRATZ: We have to generate the steam in some way. We would still have to generate steam with the electrical method of testing. Since nothing is accomplished by testing radiators in any other way than that in which they are normally used, they must be tested with steam inside. Therefore we have to have the steam. We met with some difficulties in calibration and checking up on the heat loss between the electrical boiler and the radiator, and while probably there is no particular reason for it, we simply find the condensation method more convenient and accurate.

MR. BARKER: I have used the steam method and my objection to it is that I found extreme difficulty in determining exactly when the steam was precisely saturated and didn't carry any trace of water. I used to superheat the steam by means of a gas oven and even so the small difference between the degree of saturation or superheat of the steam made a great difference in the result. I, therefore, always distrusted results obtained by that method. Do you find any difficulty in determining the degree of saturation of the steam?

PROFESSOR KRATZ: We superheat the steam about 1 deg with an electric heater placed before the radiator and we have had no occasion to distrust our results. When we are sure that we have the external conditions duplicated, we can duplicate the steam condensation very closely, and, weighing in 10-min intervals. we can duplicate the steam condensation increments well within 1 per cent as a rule. We discard any results that do not duplicate within 2 per cent.

There are arguments to be advanced on both sides, but since we have the technique developed for the condensation method, the only answer I can give you is that we find it more convenient to continue to do it that way.

F. D. MENSING: May I inquire what the temperature was in the space below the room and the space above and why those temperatures were maintained?

PROFESSOR KRATZ: The temperature in the space above the room was 62 F. We maintain such a temperature there because that is practically what is obtained on the floor or near the floor of an intermediate story, or of a room above a room in an intermediate story. There is considerable doubt as to just what temperature conditions are underneath a floor, and we aim to run our tests so that we have practically no heat flow through the floor. In order to accomplish this, we have thermocouples on the top surface of the floor and the lower surface of the floor, and maintain a temperature difference of about 1 deg between the readings of the couples on the top surface and those on the lower surface.

No. 928

A STUDY OF THE COMBUSTIBLE NATURE OF SOLID FUELS

By R. V. Frost,[1] Norristown, Pa.
MEMBER

Many consumers of fuel have experienced the trouble that accompanies the use of a fuel having a variable burning characteristic. "Variable burning" in the sense in which it is used in this paper, expresses a new thought in the study of the combustible nature of solid fuels. The phrase has not been employed by other research agencies in their reference to the subject but because of the different angle of approach outlined in this paper a different phraseology seemed essential for its proper expression.

WHILE the difficulty experienced with fuel having a variable burning characteristic is common to any fuel—solid, liquid or gaseous—the trouble has, from a practical standpoint, been virtually eliminated in the burning of liquid and gaseous fuels and coke. But in the use of natural coals, it remains one of the most serious and at the same time, the least understood of all the troubles with which the coal industry is burdened. It is a trouble common to both bituminous and anthracite coals and because of the increasing use of automatic burning apparatus, it is daily becoming of more vital consequence to the coal industry.

Many agencies are now endeavoring to obtain a solution of the problem. The U. S. Bureau of Mines has been engaged in a study of it for years, so far as it affects bituminous fuel, and every coal producer who boasts a research laboratory has delved into the subject. The statement has been made many times that coal will never "come into its own" until it is sold in liquid form, and many coal operators are today seriously considering the advisability of turning their properties into gas plants and transporting their product through pipe lines in the form of gas. Such is the importance of this subject in the eyes of those best informed.

---

[1] President, Frost Research Laboratory.

Presented at the 38th Annual Meeting of the American Society of Heating and Ventilating Engineers, Cleveland, Ohio, January, 1932.

Variability in the burning characteristics of coal makes itself evident in variation in the intensity of draft required to maintain a stated rate of combustion, and in the variability with which the fuel will ignite. The burning characteristics of coals are roughly classified as either hard or free burning, but there is no means at present in use by which a coal may accurately be classified as either hard or free burning. Appearance, while commonly used as such, is not an indicator. Neither are hardness, density, chemical analysis, location in the mine area or simple burning tests. Location is perhaps the best indicator, but this is not by any means positive, for frequently samples taken from the same lump are absolutely opposed in their burning characteristics. Sometimes the properties of a coal are so variable that its burning characteristics will change from one shovel to the next, and it is very difficult under the present methods of mining and preparation to obtain coal that is not more or less variable. In many cases, the variability is so extreme that of two samples from the same car compared on the basis of equal draft intensity, one will burn at a rate of combustion 5 or 6 times that of the other.

It is easy to imagine what a serious problem such variability in fuel becomes to manufacturers of boilers and automatic coal burning devices. This variability is undoubtedly the cause of many of the complaints raised regarding so-called inadequate boiler ratings and poor chimneys, in which case the boiler manufacturer becomes a party to the complaint. In the stoker industry the seriousness of variable burning characteristics is generally recognized and is regarded with genuine concern.

The correcting of the variability of burning characteristics is of more vital importance to the coal industry than is the production of a low ash coal, and of equal importance to uniform sizing, at least in the anthracite territory. In fact, a greater variability in sizing of anthracite is permissible than of variability in burning characteristics.

On account of combustion troubles encountered in the course of research in the operation of automatic coal burners, the writer and his associates began a study of this subject a year or more ago. Realizing its importance to the successful development of the stoker industry, the laboratory staff was led to conduct an investigation on the combustibility of all types of solid fuels. The efforts in this direction were rewarded with success in the development of a practical solution of the problem so far as anthracite and coke are concerned, but the method devised could not be applied with success to coking fuels.

In developing the method, an effort was made first to determine accurately how to classify the burning characteristics of a fuel and then to devise a method by which these characteristics could properly be indicated. Charts showing the relationship between draft and rate of combustion and the relationship between rate of horizontal ignition or fire travel and rate of combustion were devised. Several of these charts are included in this paper. (See Figs. 1 to 12, inclusive.)

BLENDING OF COALS

The final step was to devise a system by which the practical use of the process could be made. In doing this, the laboratory determined the result of blending coals of different burning characteristics. Finding that blending overcame all the difficulties of variability, a method was developed by which the

coal producer could so control his production that he could produce a composite coal so blended that its burning characteristics would continue with undisturbed uniformity from hour to hour, from day to day and from month to month, so long as he maintained an intelligent laboratory control of his production.

The first step in this research, which resulted in a laboratory system of

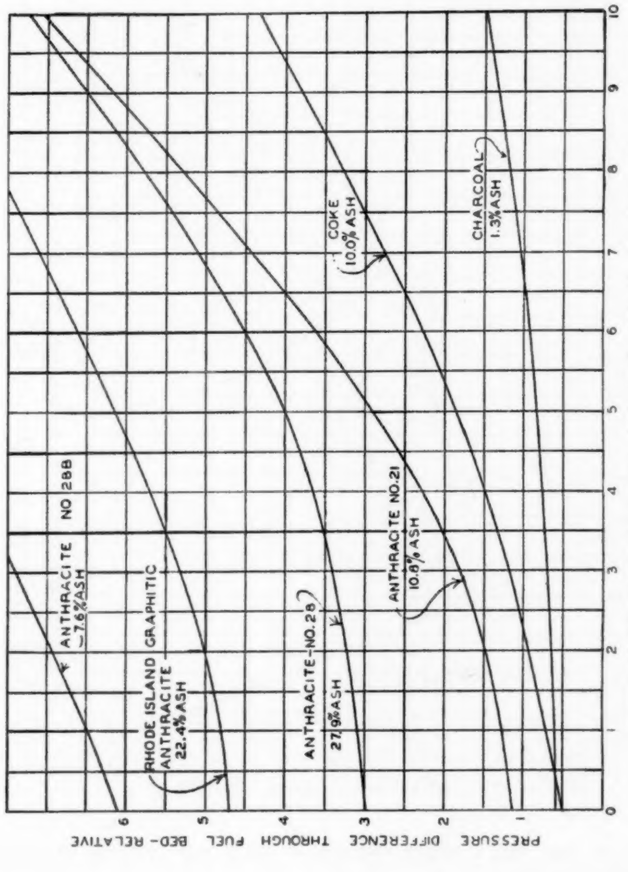


Fig. 1. Relationship Between Draft and Rate of Combustion for Samples of Several Different Fuels

testing, and the final step which is a matter of concern only to the coal producer, are omitted from further discussion in this paper. But the second step in the research, namely, the chart system showing the relationship of draft, rate of combustion and rate of ignition, together with the bearing of the entire problem upon the subject of heating, is a matter of interest to heating engineers.

Fig. 1 shows the relationship between draft and rate of combustion for samples of several different fuels. The lowest line on the chart represents a charcoal sample. This fuel required but little draft to maintain combustion and

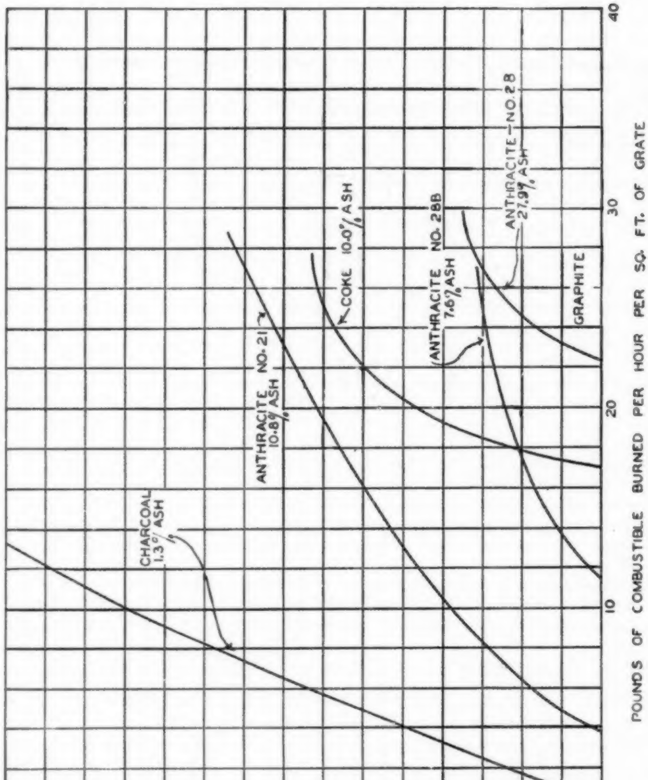


Fig. 2. Relationship of Rate of Horizontal Fire Travel and Rate of Combustion

it is apparent that only a slight increase in draft was required to produce a high combustion rate.

Coke, which had a slightly higher resistance to high combustion rate, comes next on the scale. Then appear two anthracites, and then a sample of Rhode Island anthracite, a coal which offers so high a resistance to combustion that it is not offered for sale as a fuel. Yet this supposedly valueless anthracite apparently will burn much more readily than pure Pennsylvania anthracite, the

curve of which appears at the extreme upper left corner of Fig. 1. This sample (No. 28*B*) is of unusual interest because it was a sample picked from a lump contained in sample No. 28, the curve of which is also shown. While the

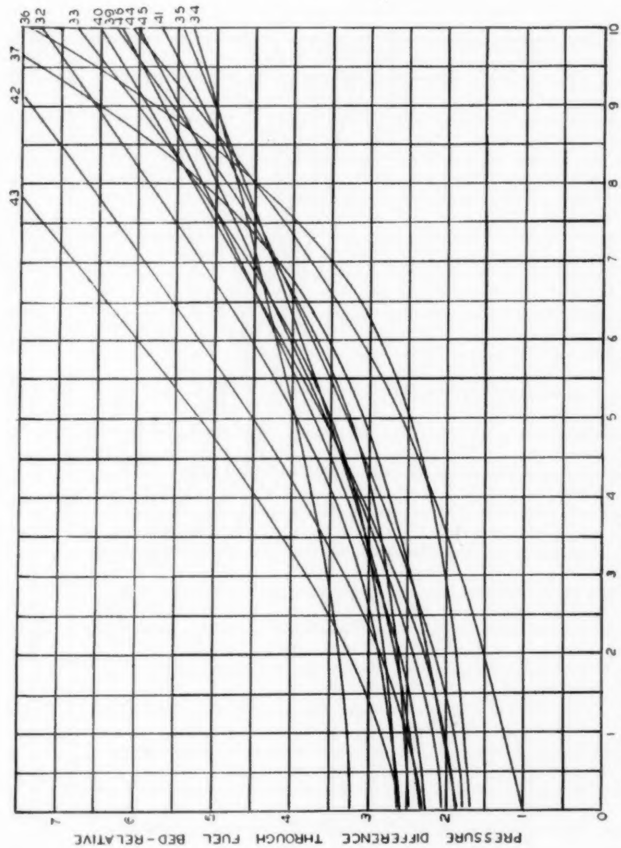


FIG. 3. CURVES SHOWING RELATION BETWEEN RELATIVE DRAFT PRESSURES AND RATE OF COMBUSTION FOR SAMPLES TAKEN FROM EACH VEIN MINED AT *V* COLLIERY

average ash of the entire No. 28 sample was 27.9 per cent as in the mine—this being a sample representing a certain location in a mine—sample No. 28*B* was selected from the whole as one representing a coal of excellent appearance and low ash content. Nevertheless, the burning characteristic of this fine appearing, low ash coal was inferior to the so-called valueless Rhode Island fuel.

Coal similar to sample No. 28*B* would require an exceptional draft to burn it, for by examination of the curves it is seen that sample No. 28, of which

No. 28*B* is a part, will burn at rate of combustion of 9 lb per square foot of grate area per hour under a draft intensity which would not permit No. 28*B* to burn at the rate of even 1 lb per hour. It is also of interest to note that sample No. 21 is an anthracite that burns at low rates with as little draft as

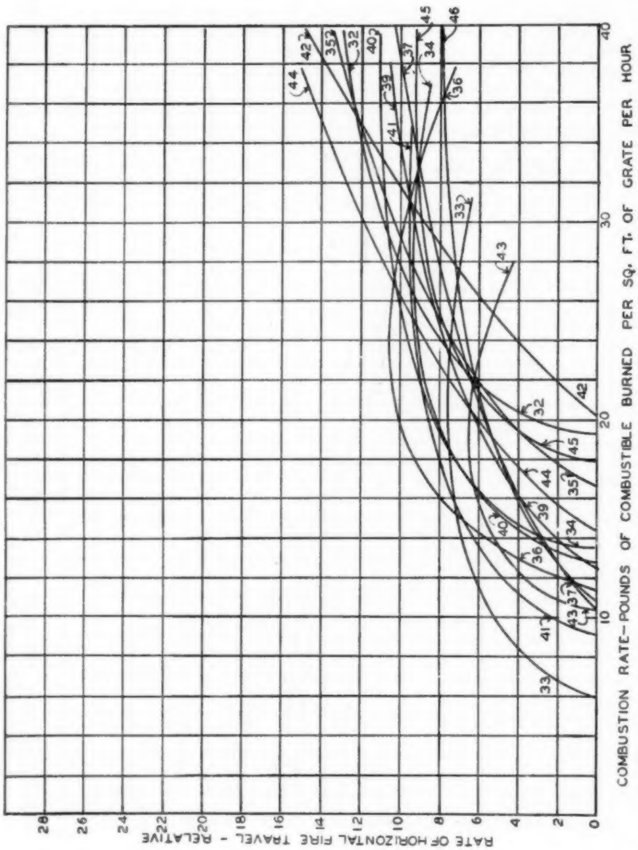


Fig. 4. Curves Showing Relation Between Relative Rates of Horizontal Fire Travel and Rates of Combustion for Samples Taken from Each Vein Mined at *V* Colliery

does coke and further that a coal of 28 per cent ash as represented by No. 28 burns almost as readily as a coal of 10 per cent ash as represented by No. 21.

Horizontal Ignition

Fig. 2 shows the relationship between rate of horizontal fire travel and rate of combustion. The ***rate of horizontal fire travel*** as used in this paper is an

index of the rapidity with which the fire will spread from one part of the fuel bed to another. It is useful to indicate the rate of ignition and a coal of quick ignition is of course a free-burning coal.

In Fig. 2, charcoal again appears superior but coke is seen to be slower of

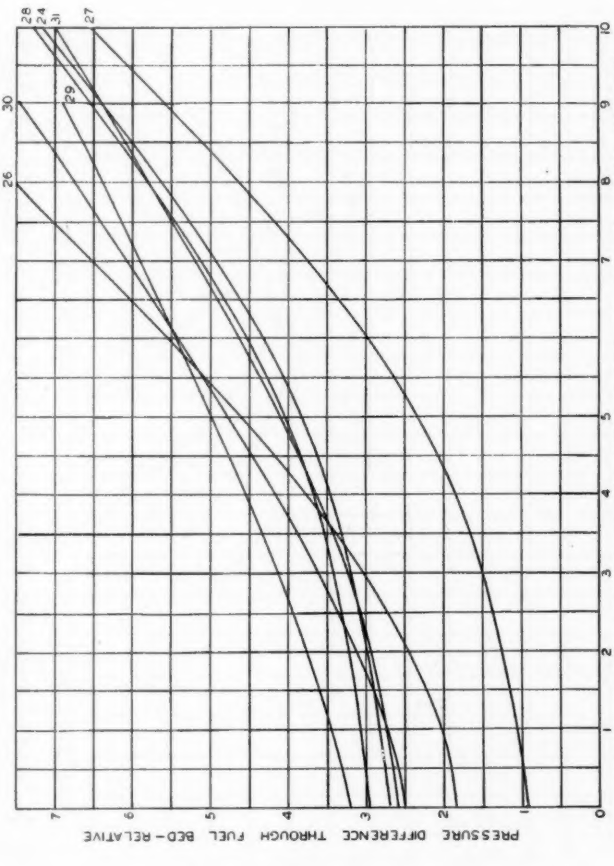


FIG. 5. CURVES SHOWING RELATION BETWEEN RELATIVE DRAFT PRESSURES AND RATE OF COMBUSTION FOR SAMPLES TAKEN FROM EACH VEIN MINED AT *W* COLLIERY

ignition than anthracite No. 21, or even the anthracite of high draft requirement (No. 28*B*), while sample No. 28 falls to the lowest because of its high ash content. From these comparisons, it might be inferred that the ash particles act as an insulator against rapid ignition since on this chart the positions of the high ash samples No. 28 and the low ash No. 28*B* are reversed with respect to Fig. 1. Furthermore, Rhode Island graphite is found to have no horizontal ignition properties at all, thus barring it from use as a fuel. Coke, while having

a slow initial horizontal ignition quickly overcomes the lead due to its greater proportion of voids.

In order to reflect the combustible nature of anthracite generally, samples were obtained from 5 different anthracite mines during the course of this in-

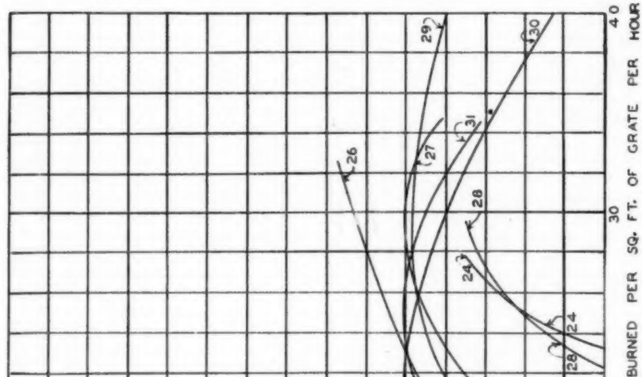


FIG. 6. CURVES SHOWING RELATION BETWEEN RELATIVE RATES OF HORIZONTAL FIRE TRAVEL AND RATES OF COMBUSTION FOR SAMPLES TAKEN FROM EACH VEIN MINED AT *W* COLLIERY

vestigation, the samples being so selected as to represent a general survey of the coal mined in each property. The graphs of the combustibility tests on the samples from three of the mines are shown in Figs. 3 to 8, inclusive.

The curves for the survey at colliery *X* (Figs. 7 and 8) show a very close uniformity between the samples from the various veins. This mine is favor-

ably known because of the uniformity and free burning characteristics of its product.

Collieries *V* and *W* charted on Figs. 3 to 6 are similar to the great majority of anthracite mines and illustrate the general range of variability in this fuel.

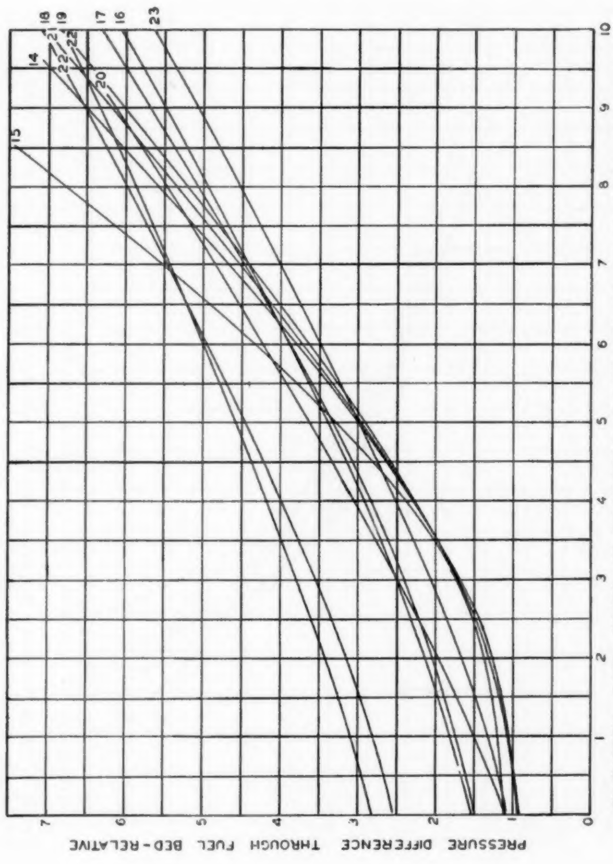


FIG. 7. CURVES SHOWING RELATION BETWEEN RELATIVE DRAFT PRESSURES AND RATE OF COMBUSTION FOR SAMPLES TAKEN FROM EACH VEIN MINED AT X COLLIERY

Colliery *W* is perhaps more troublesome in combustible characteristics because there are fewer veins and these few widely distributed.

The method of production now in vogue at these mines does not permit of blending except perhaps in an accidental way. When a train of mine cars is hauled from the mine to the top of the breaker there may be 10 cars all

loaded at one locality in the mine. This coal passes through the breaker, is washed and sized and loaded into a railroad car entirely segregated from the coal from any other part of the mine. The coal loaded in a car may thus be either of the hardest or the freest burning coals or a hard burning coal may

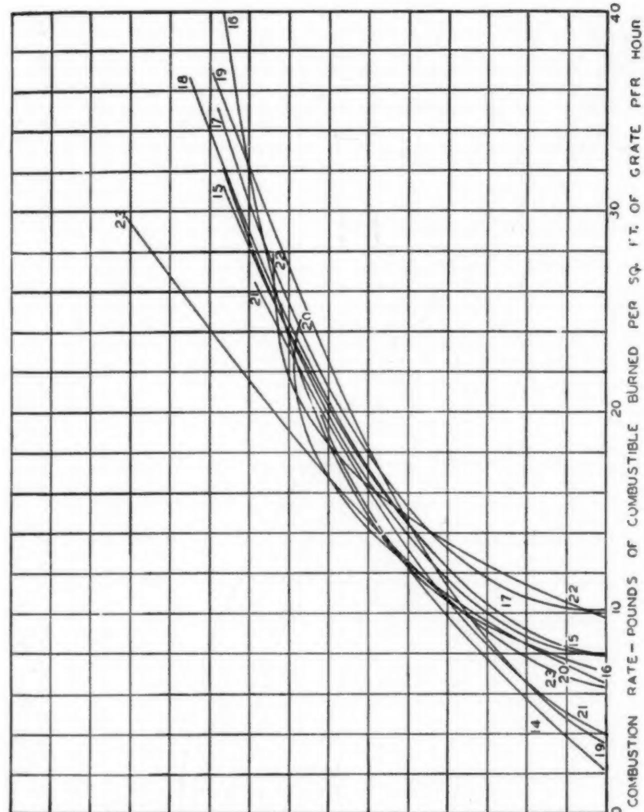


FIG. 8. CURVES SHOWING RELATION BETWEEN RELATIVE RATES OF HORIZONTAL FIRE TRAVEL AND RATES OF COMBUSTION FOR SAMPLES TAKEN FROM EACH VEIN MINED AT X COLLIERY

be deposited in the car followed by a free burning coal, resulting in a conglomerate mixture not uniformly blended.

That part of a consumer's heating equipment comprising the heater and chimney may not operate economically on a free-burning coal, or the plant may have an improperly-designed chimney, thus requiring a free-burning coal,

or again the consumer may find he has received an unbalanced mixture, resulting first in a free-burning fire and then in a slow-burning fire. The outcome is annoyance and dissatisfaction, followed by a complaint to the dealer and transmitted by the dealer to the producer. Neither the dealer nor producer,

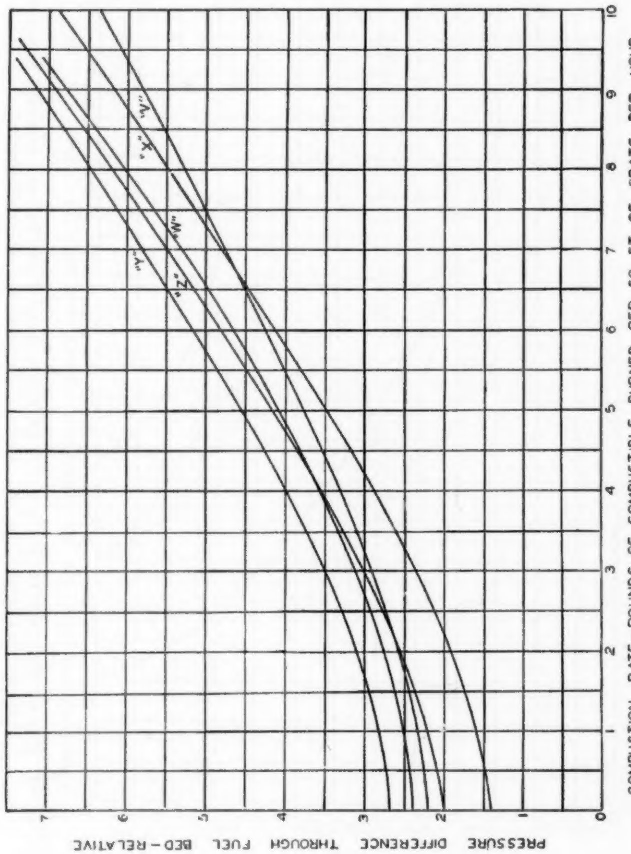


FIG. 9. RELATIVE DRAFT PRESSURES PLOTTED AGAINST RATE OF COMBUSTION FOR AVERAGE SAMPLES FOR EACH LOCALITY

having records of the combustible character of the fuel, is able to determine the cause of the trouble and continues in ignorance, repeating the same offense over and over. Clinkers, sizing, bone and slate and ash percentage are all blamed, whereas if the coal had been uniformly blended no complaint would have been made.

That proper blending will result in a very uniform coal is apparent from Figs. 9 and 10 which show curves of blended coals from the 5 different mines. These mines spread from one end of the anthracite field to the other, yet when the coals from each mine are properly blended the composite coal from any

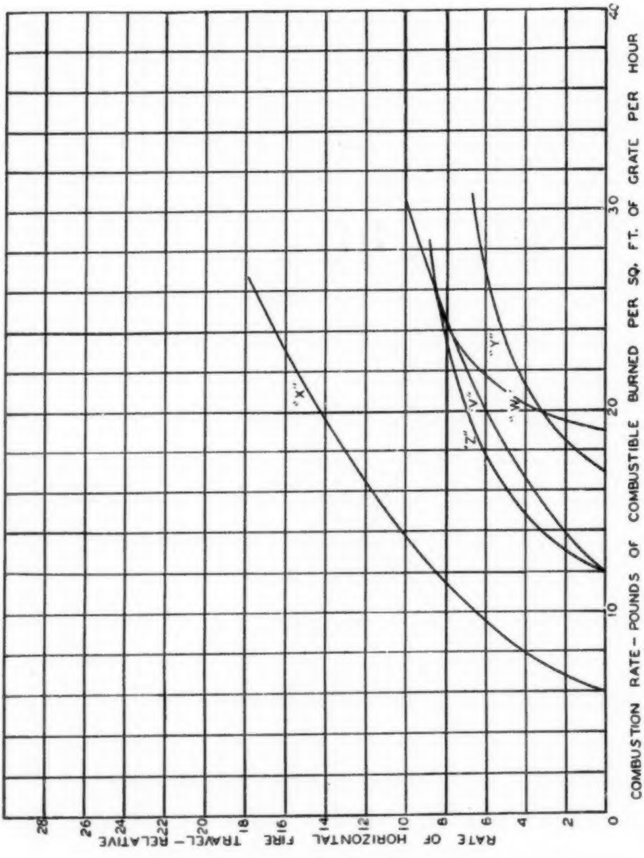


FIG. 10. CURVES SHOWING RELATIVE RATES OF HORIZONTAL FIRE TRAVEL PLOTTED AGAINST RATES OF COMBUSTION FOR AVERAGE SAMPLES FOR EACH LOCALITY

one mine is very similar in combustible character to the composite coal from any of the other 4 mines, the single exception being from mine *X*, where the coals are so uniform in character that the blend is still much freer burning than the blended coal from any of the others. Not only does the blend make a more uniform coal but it makes a coal that requires less draft, ignites more readily, and produces less clinker.

The reason the blended coal burns with less clinkers is explained by the fact that the curves of the coals having clinkering tendencies take a decided upturn at some point on the chart. When this takes place it indicates that the temperature of the fuel has reached a point that the ash sinters, thus closing off

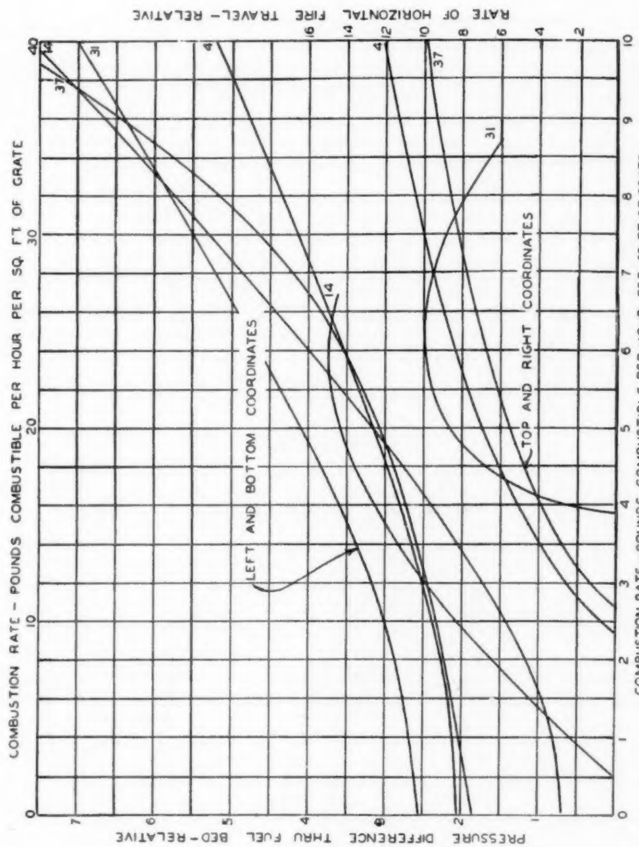


FIG. 11. COMBUSTIBILITY OF COAL VEIN *B* AT DIFFERENT LOCALITIES

the admission of air to the carbon. In the blended coal the mixture of ash from the various veins tends to prevent the sintering of the various particles in the ash, perhaps due to the insulating effect of the non-clinkering ash which keeps the particles likely to fuse from reaching a critical fusion temperature.

Furthermore, it is clearly demonstrated that the ash structure is responsible

for either the free or hard burning characteristic of a coal. This is explained by the fact that in the free burning coal the ash falls away from the carbon as it is formed thus permitting contact of the hot carbon with the oxygen of the air, whereas if the ash retains its form and continues to enclose the carbon the fuel is hard or slow burning. Observation has shown that it is

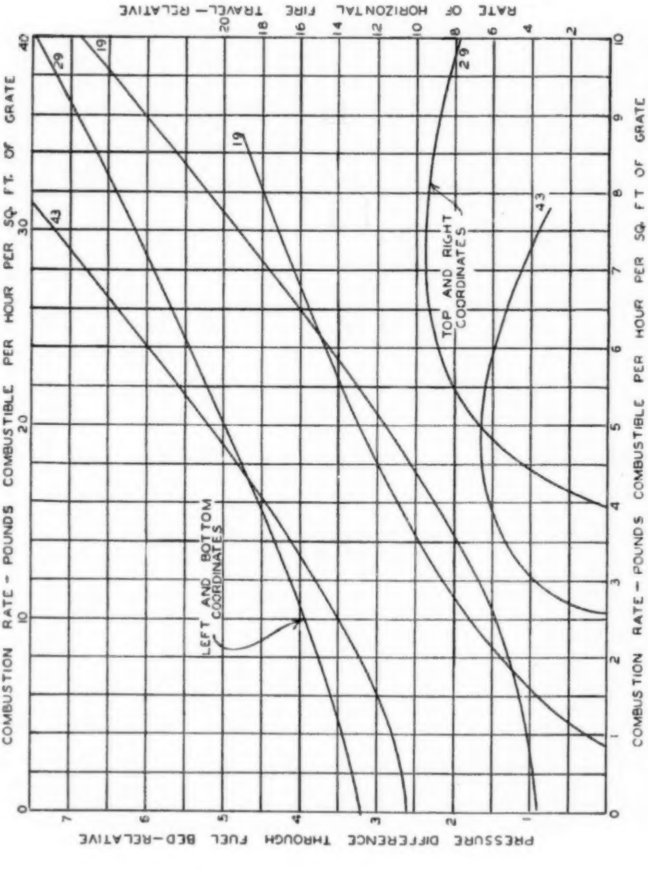


FIG. 12. COMBUSTIBILITY OF COAL VEIN *F* AT DIFFERENT LOCALITIES

very difficult to ignite the carbon through an ash exceeding one-half inch in thickness thus explaining why the coals of the fine prepared sizes can be burned to a more perfect ash than can the so-called senior or domestic sizes.

Another phase of the surveys of the 5 mines is shown in Figs. 11 and 12. It is frequently claimed that coal from a particular vein is of unusual quality

regardless of the section of the field in which the vein is located. That this claim is unfounded is illustrated by the curves of the same veins as they were found in the different mines. The wide spread of the graph lines shows the wide divergence in the combustible character of the fuel from the different areas. Coal from a certain vein in one mine may be very free burning while coal from the same vein in another mine may be extremely hard burning.

CONCLUSIONS

It is apparent that the usual tests of coal fail when applied to the determination of the combustible character of the coal. Neither appearance, density, chemical analysis nor location can be used as an indicator. The only positive means of determining the combustible character of a coal before it is put to use is to make a specific test for that particular purpose.

That the determination of the combustible character of the coal is highly essential to its proper utilization cannot be denied. Under hand-firing practice it was possible to blunder along with efficiencies of 50 or 60 per cent without serious complaint but with the increasing employment of automatic burning devices and the increasing use of coke, oil and gas as competitive fuels a coal that is not uniform in combustible character becomes a drag upon the entire practice of coal utilization, for with the absolute necessity of close adjustment and balance between air and coal feed in the automatic burning of coal, any coal that is not uniform in combustibility is immediately condemned.

DISCUSSION

F. W. HANBURGER (WRITTEN): I now know that heretofore I had never fully appreciated the importance of the *variability* of the burning characteristics of fuels, to use Mr. Frost's term.

The graph showing the uniform combustion characteristics and non-clinkering qualities of blended coals is very interesting, and should prove a great aid to the automatic coal stoking industry. I understand that the clinkering of coals causes fissures in the bed in the fire pot of various stokers, allowing a too free air passage, lowering the combustion efficiency and carrying dust to the roof.

From Mr. Frost's paper it is evident that this could be entirely avoided by proper blending. It may, however, be a difficult matter to determine how this blending might be accomplished.

PROGRAM SEMI-ANNUAL MEETING 1932

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

HOTEL PFISTER, MILWAUKEE, WIS.

JUNE 27, 28 AND 29, 1932

(All events on Central Standard Time)

Monday, June 27

8:30 A.M. Registration, Hotel Pfister (7th Floor)

10:00 A.M. Technical Session (Fern Room)
Greeting—E. A. Jones, President, Wisconsin Chapter
Response—Pres. F. B. Rowley
Report of the Council
Paper—Velocity Characteristics of Hoods Under Suction, by J. M. DallaValle
Paper—Natural Wind Velocity Gradients Near a Wall, by J. L. Blackshaw and F. C. Houghten
Report—Committee on Revision of Constitution and By-Laws—W. T. Jones, Chairman

12:00 NOON Ladies leave Hotel Pfister for Luncheon and Bridge at Milwaukee Yacht Club

12:30 P.M. Golfers leave Hotel Pfister for Luncheon at Ozaukee Country Club

1:30 P.M. Tournament (Research Cup)—18 Holes Medal Play

2:30 P.M. Visit to A. O. Smith Laboratories

6:30 P.M. Committee on Research—Dinner-Meeting

8:30 P.M. Entertainment, Dance, Dutch Lunch (9th floor Wisteria Room)

Tuesday, June 28

9:30 A.M. Technical Session (Fern Room)
Report—How to Use the Effective Temperature Index and Comfort Charts (Report of Technical Advisory Committee on Re-Study of Comfort Chart and Comfort Line, C. P. Yaglou, Chairman)
Paper—Carbon Monoxide Distribution in Relation to Garage Ventilation, by F. C. Houghten and Paul McDermott
Paper—Investigation of Air Outlets in Classroom Ventilation, by G. L. Larson, D. W. Nelson and R. W. Kubasta
Report—Committee on Ventilation Standards—W. H. Driscoll, Chairman

10:30 A.M. Ladies Auto Trip and Luncheon (Lake Shore, Parks, Conservatory, Sunken Gardens, Zoo)

12:30 P.M. Luncheon Nominating Committee—H. M. Hart, Chairman

12:30 P.M. Cars leave hotel for Luncheon and Golf at North Hills Country Club

2:30 P.M. Inspection of Industrial Plants

7:30 P.M. Semi-Annual Banquet and Dance (Fern Room, Hotel Pfister)

Wednesday, June 29

9:30 A.M. Technical Session (Fern Room)
Report—Chapter Relations Committee—E. K. Campbell, Chairman
Paper—Thermal Properties of Building Materials, by F. B. Rowley and A. B. Algren
Paper—Tests of Convectors in a Warm Wall Testing Booth, by A. P. Kratz, M. K. Fahnestock and E. L. Broderick
Paper—Loss of Head in Copper Pipe and Fittings, by F. E. Giesecke and W. H. Badgett
Paper—Automatic Gas Burners, by C. G. Segeler
Adjournment

10:30 A.M. Ladies' Choice—Golf, Museum, Beaches or Inspection of Pfister Art Gallery and Period Rooms.

No. 929

SEMI-ANNUAL MEETING, 1932

AN EXTREMELY busy three days were enjoyed by the members of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, who gathered in Milwaukee, Wis., June 27-29, for the Semi-Annual Meeting 1932. The registration totalled 335, the largest turnout recorded in several years. In addition to the presentation of 11 technical papers, two items of outstanding interest were the Report of the Committee on Ventilation Standards, which was referred to the entire membership for acceptance by letter ballot, and the Proposed Revisions to the Constitution and By-Laws, which will be sent to the members for comment in advance of the next Annual Meeting.

The Council accepted the cordial invitation of the newly chartered Cincinnati Chapter to hold the Annual Meeting in January 1933 in Cincinnati.

The first session of the Semi-Annual Meeting 1932 convened at the Hotel Pfister, Milwaukee, Wis., on Monday morning, June 27, with President F. B. Rowley presiding.

E. A. Jones, President of the Wisconsin Chapter, gave a brief address of welcome in which he expressed the pleasure of the local members for the opportunity of entertaining the Society and the desire that everyone would have a pleasant and profitable visit in Milwaukee.

John S. Jung, Chairman of the Committee on Arrangements, was introduced and made several announcements relating to the entertainment program which had been arranged for the pleasure of the visiting members and ladies.

President Rowley called upon W. T. Jones for a report of the Committee on Revision of Constitution and By-Laws. Mr. Jones stated that the report of the committee had been printed and it was not the intention to consider the matter in detail, but merely to call the attention of the membership to the fact that the committee had been working for the past two years in an effort to revise our Constitution and By-Laws and the various rules and regulations, so that there would be no conflict and references would be easy to find. The whole subject was divided into three separate sections: (1) Constitution, (2) By-Laws and (3) Rules. Each division has corresponding sections and titles, so that all information on funds which appear in the three divisions can be located under the same number.

It is the suggestion of the committee, Mr. Jones explained, that this report as submitted to the meeting, be sent out to the entire membership with the approval of the Council, so that final amendments can be presented at the Annual Meeting next January. After the Annual Meeting, the Constitution, By-Laws and Rules will then be voted upon by letter ballot. This procedure

will make the new Constitution effective in a year, although it is readily understood that the matter could be discussed and final approval delayed for the next 10 years. In addition to the Constitution and By-Laws, the Regulations Governing the Committee on Research will become a part of the Rules.

The following motion was then presented by Mr. Jones:

Resolved, that the revised draft of the Constitution, By-Laws and Rules and the Regulations of the Committee on Research, after approval by the Council, shall be sent to the entire membership of the Society and then presented at the next Annual Meeting.

The motion was seconded by J. D. Cassell and was unanimously carried.

During the second session President Rowley introduced E. K. Campbell, Chairman of the Technical Advisory Committee on Ventilation of Garages, who also represented the Society on the Garage Committee of the *National Fire Protection Association.* He explained that work has been carried on by the staff of the Society's Research Laboratory in various garages and two papers had been prepared for presentation. Mr. Campbell also announced that the *National Fire Protection Association* had been developing regulations for garages from the standpoint of fire protection and the Society's Code on Garage Ventilation had been approved by the *N. F. P. A.* with some minor changes. Since the completion of the code work, he said, the Technical Advisory Committee had carried on some investigations in St. Louis, in Pittsburgh and later at Lawrence, Kansas in cooperation with the University of Kansas. The present reports cover the work done at Pittsburgh.

Report of Committee on Ventilation Standards [1]

President Rowley stated that the next report from the Society's Committee on Ventilation Standards was of exceptional interest to the members. The report presented at the Annual Meeting in Cleveland last January had been thoroughly discussed and the suggestions made had been considered by the committee in preparation of the present report. In the absence of Chairman W. H. Driscoll, Prof. A. C. Willard read the statement which had been prepared by Mr. Driscoll.

Chairman's Discussion of the Report

Circumstances which have made it necessary for me to hurriedly change my plans with respect to my attendance at the Society meeting seem to impose on me the necessity of placing before the Society certain views that have been formulated as a result of my experience as Chairman of the Committee on Ventilation Standards.

In this connection, the following is submitted as a discussion of the report to be presented at the Tuesday morning session.

I hope that I may be pardoned if, in my hurry to get this statement off in the earliest possible mail, I may repeat some statements that have heretofore been made on the subject.

The purpose of the committee is not, as some seem to think, to frame a bill for enactment into law in the different states. Its duty is to set up clear and definite standards of ventilation representative of what it believes to be the thought of the

[1] For Text of Report see p. 383.

Society on this important subject. These standards, when and if adopted by the Society, may be used by individual members, or chapters, or others interested in promoting laws, building codes or health regulations, with the assurance that they have back of them the support and authority of the A.S.H.V.E.

The Society, as a national body, does not propose to concern itself with the promoting or passage of such laws. It is, however, trying to perform a long neglected duty, *i.e.*, the setting up and sealing with its approval of such standards of ventilation as it believes in the light of its experience, its study, and its research activities, properly serve the requirements of human health and comfort.

Your committee has never been optimistic enough to feel that it would emerge from its deliberations with a perfect instrument; nor, on the other hand, does it agree with a few members of the Society who feel that, because this is so, the Society should stand idly by and do nothing.

In considering this subject, the members should bear in mind that the standards of ventilation, like any other standards of the Society, are not necessarily permanent, but are subject to change to meet new conditions and to conform to a more enlightened understanding. This should dispose of the fear expressed by one or two that if we set up a definite air volume of 10 cubic feet we may find in the future that this may be reduced to nine or five, or may be omitted entirely.

Some members have stated that the establishment of ventilation standards lies only within the realm of the physiologist and that the engineer, therefore, should calmly fold his hands and wait for instructions. These members lose sight of two important facts:

First—That the medical profession has wrestled intermittently with the problem for a century and a half, and has yet to produce standards that will serve either the needs or purposes for which our standards are intended.

Second—It is surely the function of the engineer to provide the means for controlling the conditions that are demanded for health and comfort, and it is certainly within his province to work out the engineering problems of air volume, distribution and circulation required to maintain comfort conditions.

The writer has endeavored to keep before the members of the committee at all times the arguments of those who do not favor the inclusion of the definite air volume requirement as set forth in Section V of the report. This seems to be the only real point on which any real issue has been raised. The only difference of opinion in the committee on this issue has been expressed in writing by one member, who incidentally is the only member who has never attended a committee meeting. We have been very particular, however, to place in the hands of every member of the committee a copy of his communication and to give every consideration in and out of the meetings to the views expressed therein.

With this single exception, the committee has been unanimous in its belief that the 10 cubic feet should be retained in the report as written. This belief is based on the statements of those members of the committee, and of the Society, who have had experience in the ventilation of places of assembly and is supported by the investigations made by Mr. Yaglou at Harvard. Furthermore, support for this figure is found in the *Congressional Record*, January 5, 1928, Pages 1067 and 1068. The subject matter of this article is the ventilation and air conditioning of the legislative chambers of the United States Capitol. It contains a report of the committee appointed by the United States Public Health Service to investigate the matter. The chairman of this committee was Professor C.-E. A. Winslow. The report of this committee recommends a minimum of 10 cubic feet of air per person per minute, based on the maximum occupancy of the chambers.

In conclusion, would advise that the writer has been endeavoring to obtain from the medical directors of the large insurance companies of the United States their opinion on this subject. It has been difficult to obtain direct or definite statements because, in general, they do not consider it politic for insurance companies to enter into a discussion of a controversial subject. I take the liberty, however, of quoting the following from one reply that I received:

> "It is well to remember that medicine is an art as well as a science, and while doctors prefer to base their recommendations on scientifically demonstrated facts,

nevertheless, much of our therapy is based simply on what we have learned from experience. No doctor, for instance, knows why the salicylates have proven useful in rheumatism or why mercury should be one of the sovereign remedies for the treatment of other diseases, and yet experience has taught that these remedies are valuable. So no physician knows why fresh air is so important in the treatment and prevention of tuberculosis and in the maintenance of good health in general. We simply know that experience has taught us that this is true."

It seems to me that, in view of the preponderance of opinion in the medical profession of the value of outdoor air and despite the fact that they have never scientifically demonstrated the necessity for a definite quantity per person, that they will not disregard the broad experience of the members of this Society. I am convinced that the medical profession generally will willingly accept the 10 cubic feet as a satisfactory volume until it has been scientifically demonstrated or determined that this volume is not correct. The burden of proof, however, it seems to me, is on the shoulders of those who take exception to this figure, inasmuch as, in the opinion of the committee, it clearly represents the Society's viewpoint.

I am heartily in favor of the adoption of the report as written and sincerely hope that the Society takes such favorable action at the earliest possible moment.

President Rowley announced that the subject was open for discussion and S. R. Lewis, Chicago, said it seemed that action on this report should be taken by the membership of the Society after some deliberation, preferably by letter ballot, and made the following motion:

That the Report of the Committee on Ventilation Standards be accepted and transmitted to the entire voting membership of the Society, for "yes" or "no" acceptance.

The motion was seconded by H. M. Hart, Chicago, and on vote was unanimously carried.

Upon inquiry of J. J. Aeberly, Chicago, President Rowley stated that the process of mailing the Code for vote of the membership would be speeded up insofar as possible, in order that the members could express their opinion promptly.

The final session of the Semi-Annual Meeting, 1932, was opened by President Rowley, who introduced E. K. Campbell, Chairman of the Committee on Chapter Relations. Mr. Campbell gave a brief outline of the committee's efforts to assist the local chapters of the Society with their meeting programs, arrange for speakers and promote closer relationship between the individual chapters. This committee, he indicated, acts as a general clearing house to aid any chapter and welcomes the ideas of members and officers of these organizations.

The report of the Nominating Committee was presented by H. M. Hart, Chicago, Ill.

Resolutions

The following resolution was offered by Mr. Hart and seconded by J. D. Cassell:

Whereas, Mr. George Mehring of Chicago was a Charter member of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, and

Whereas, he was a member of the group which formed the Illinois Chapter, and

Whereas, its President in 1909, and

Whereas, he did much to aid in the development of the profession and the industry in which he was always a leader, and

Whereas, his spirit has passed on to eternal life;

Be It Resolved, that we, his associates, do bow our heads in respect for the departed friend; and

Be It Further Resolved, that this expression of friendliness and regret be spread upon our records and that a copy be forwarded to the widow and family of the man we knew and revered.

W. W. Timmis, New York, offered the following resolution which was seconded by Mr. Cassell and adopted:

Resolved, that inasmuch as the lower limit of the relative humidity for comfort and health has not been established beyond reasonable question by the research either of this Society or of others; it is the sense of this meeting that the Committee on Research be instructed to conduct research looking toward the determination of such lower limit as soon as practicable, and to report its findings to an Annual or Semi-Annual Meeting of the Society.

Prof. G. L. Larson, Chairman of the Committee on Research, remarked that a study of a lower humidity limit had been planned and would undoubtedly be carried forward as soon as funds were available. He pointed out it was difficult to undertake new work unless special funds were directly provided for this particular investigation.

Prof. A. C. Willard, Urbana, said that Mr. Timmis' resolution was a fine exhibition of the proper spirit and attitude toward the difficult situation in which the Society will constantly find itself involved in the preparation, adoption and perfection of its various codes. The author of the resolution has reasonable doubt about the humidity limit but he does not wish to delay the adoption of a code which is substantially satisfactory and he suggests a reasonable way of obtaining the answer to his question.

Report of Resolutions Committee

Mr. Cassell presented the report of the Resolutions Committee as follows:

Resolved, that a vote of thanks and appreciation of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS be extended to the Mayor of Milwaukee and City Council for the fine hospitality and many courtesies extended.

Resolved, that a vote of thanks and appreciation be extended to the Milwaukee Association of Commerce for the services rendered and the very fine cooperation they have extended during our stay in this city.

Resolved, that a vote of appreciation be extended to the Wisconsin Chapter of the A.S.H.V.E. for their splendid reception of our members and ladies and that we express our sincere thanks for the pleasant and happy events they have provided for our entertainment during the three days of our stay.

Resolved, that a vote of thanks be extended to the management of the Hotel Pfister for the very fine service rendered our members and guests.

Resolved, that we extend the appreciation of this organization to the Technical and Public Press for the efforts they have extended in giving this Society most desirable and accurate publicity of our sessions.

As there were no items of new business offered for discussion, President Rowley announced that the registration totaled 335, far exceeding expectations, and paid a tribute to the Wisconsin Chapter and John S. Jung, General Chairman of the Committee on Arrangements, for the work that had been done in entertaining the visiting members and guests.

COMMITTEE OF ARRANGEMENTS

JOHN S. JUNG, *General Chairman*

H. W. ELLIS, *Advisory*
J. G. SHODRON, *Reception*
G. L. LARSON, *Registration*
A. M. WAGNER, *Transportation*
V. A. BERGHOEFER, *Finance*
E. A. JONES, *Golf*
F. G. WEIMER, *Banquet*
ERNEST SZEKELY, *Publicity*

C. H. RANDOLPH, *Ladies Entertainment*
H. F. HAUPT, *Entertainment*
MRS. E. A. JONES, *Hostess*

No. 930

A. S. H. V. E. VENTILATION STANDARDS

COMMITTEE

W. H. Driscoll, *Chairman;* J. J. Aeberly, F. Paul Anderson, L. A. Harding, D. D. Kimball, J. R. McColl, C. L. Riley, W. A. Rowe, Perry West and A. C. Willard.

PREAMBLE

It is the intent of the Committee in presenting this report to confine itself to a statement of those requirements which, based on present day knowledge, will provide adequate ventilation for spaces intended for human occupancy. The following standards shall apply to all spaces occupied by human beings in all buildings for which ventilation regulations are to be established.

SECTION I—AIR TEMPERATURE AND HUMIDITY

The *temperature* and *humidity* of the air in such occupied spaces, and in which the only source of contamination is the occupant, shall be maintained at all times during occupancy at an *Effective Temperature,* as hereinafter stated.

The relative humidity shall be not less than 30 per cent, nor more than 60 per cent in any case. The Effective Temperature shall range between 64 deg and 69 deg when heating or humidification is required, and between 69 deg and 73 deg when cooling or dehumidification is required.

These Effective Temperatures shall be maintained at a level of 36 in. above the floor. (See Appendix, Tables 1 and 2.)

SECTION II—AIR QUALITY

The air in such occupied spaces shall at all times be free from toxic, unhealthful or disagreeable gases and fumes and shall be relatively free from odors and dust.

In every space coming within the provisions of these requirements and in which the quality of the air is below the standards prescribed by good medical and engineering practices, due to toxic substances, bacteria, dust, excessive temperature, excessive humidity, objectionable odors, or other similar causes, means for ventilating shall be provided so that the quality of the air shall be raised to these standards.

Adopted 1932 by the American Society of Heating and Ventilating Engineers.

SECTION III—AIR MOTION

The air in such occupied spaces shall at all times be in constant motion sufficient to maintain a reasonable uniformity of temperature and humidity, but not such as to cause objectionable drafts in any occupied portion of such spaces.

The air motion in such occupied spaces, and in which the only source of contamination is the occupant, shall have a velocity of not more than 50 feet per minute, measured at a height of 36 in. above the floor.

SECTION IV—AIR DISTRIBUTION

The air in all rooms and enclosed spaces shall, under the provisions of these requirements, be distributed with reasonable uniformity, and the variation in the carbon dioxide content of the air shall be taken as a measure of such distribution.

The air in a space ventilated in accordance with these requirements, and in which the only source of contamination is the occupant, shall be distributed and circulated so that the variation in the concentration of carbon dioxide, when measured at a height of 36 in. above the floor, shall not exceed one part in 10,000.

SECTION V—AIR QUANTITY

The quantity of air used to ventilate the given space during occupancy shall always be sufficient to maintain the standards of air temperature, air quality, air motion and air distribution as herein required. Not less than 10 cubic feet per minute per occupant of the total air circulated to meet these requirements shall be taken from an outdoor source.

APPENDIX

Definitions

For the purposes of these standards the terms used shall be defined as follows:—

Ventilation: The process of supplying or removing air by natural or mechanical means, to or from any space. Such air may or may not have been conditioned. (See *Air Conditioning.*)

Air Conditioning: The simultaneous control of all or at least the first three of those factors affecting both the physical and chemical conditions of the atmosphere within any structure. These factors include temperature, humidity, motion, distribution, dust, bacteria, odors, toxic gases, and ionization, most of which affect in greater or lesser degree human health or comfort.

Dry Bulb Temperature: The temperature of the air which is indicated by any type of thermometer which is not affected by the water vapor content or relative humidity of the air.

Dust: Solid material in a finely divided state, the particles of which are large and heavy enough to fall with increasing velocity, due to gravity in still air. For instance, particles of fine sand or grit, such as are blown on a windy day, the average diameter of which is approximately 0.01 centimeter, may be called dust.

Effective Temperature: An arbitrary index of the degree of warmth or cold felt by the human body in response to temperature, humidity, and movement of the air. Effective temperature is a composite index which combines the readings of temperature, humidity, and air motion into a single value. The numerical value of the effective temperature scale has been fixed by the temperature of saturated air which induces an identical sensation of warmth.

Humidity: The water vapor (either saturated or superheated steam) occupying any space, which may or may not contain other vapors and gases at the same time.

Relative Humidity: A ratio, although usually expressed in per cent, used to indicate the degree of saturation existing in any given space resulting from the water vapor present in that space. The presence of air or other gases in the same

TABLE 1. EFFECTIVE TEMPERATURES RANGING FROM 64 DEG TO 69 DEG FOR VARIOUS DRY-BULB TEMPERATURES AND RELATIVE HUMIDITIES FOR STILL AIR FOR PERSONS NORMALLY CLOTHED AND SLIGHTLY ACTIVE [a]

(For use when heating or humidification is required)

DRY-BULB TEMPERATURES (DEG FAHR)	RELATIVE HUMIDITIES (PER CENT)						
	30	35	40	45	50	55	60
	EFFECTIVE TEMPERATURES (DEGREES)						
67						64.0	64.3
68			64.0	64.2	64.5	64.8	65.1
69	64.1	64.4	64.8	65.1	65.4	65.7	66.0
70	64.8	65.1	65.4	65.8	66.2	66.5	66.8
71	65.5	65.8	66.2	66.6	67.0	67.3	67.7
72	66.2	66.5	66.9	67.3	67.7	68.1	68.5
73	67.0	67.3	67.7	68.1	68.5	68.9	
74	67.7	68.0	68.4	68.8			
75	68.4	68.7					
76	69.0						

[a] See A.S.H.V.E. GUIDE 1932, Fig. 2, p. 394.

TABLE 2. EFFECTIVE TEMPERATURES RANGING FROM 69 DEG TO 73 DEG FOR VARIOUS DRY-BULB TEMPERATURES AND RELATIVE HUMIDITIES FOR STILL AIR FOR PERSONS NORMALLY CLOTHED AND SLIGHTLY ACTIVE [a-b]

(For use when cooling or dehumidification is required)

DRY-BULB TEMPERATURES (DEG FAHR)	RELATIVE HUMIDITIES (PER CENT)						
	30	35	40	45	50	55	60
	EFFECTIVE TEMPERATURES (DEGREES)						
73							69.3
74					69.3	69.7	70.1
75			69.1	69.5	70.0	71.5	71.0
76	69.0	69.4	69.9	70.5	70.8	71.3	71.8
77	69.7	70.2	70.7	71.2	71.6	72.1	72.6
78	70.4	70.9	71.4	71.9	72.4	73.0	
79	71.1	71.6	72.2	72.6			
80	71.8	72.4	72.9				
81	72.5						

[a] See A.S.H.V.E. GUIDE 1932, Fig. 2, p. 394.
[b] This table applies primarily to cases in which the human body has reached equilibrium with the surrounding air. A higher plane of summer effective temperatures is required in places of public assembly where the period of occupancy is short, than is required for offices and industrial plants where the period of occupancy is of longer duration. When the period of occupancy is two hours or less, the dry-bulb temperature shall be 72 F plus one-third of the difference between the outside dry-bulb temperature and 70 F, and the relative humidity shall not exceed 60 per cent.

space at the same time has nothing to do with the relative humidity of the space, which depends merely on the temperature and partial pressure of the vapor.

Spaces in Which the Only Source of Contamination Is the Occupant: Spaces in which the atmospheric contamination results entirely from the respiratory processes of the occupant, including heat, moisture, and odors given off by the body. No manufacturing or industrial processes or other sources of atmospheric contamination, including heat and moisture, than people are considered under this title.

No. 931

VELOCITY CHARACTERISTICS OF HOODS UNDER SUCTION

By J. M. DallaValle,[1] Cleveland, Ohio

NON-MEMBER

EVERY hood under suction creates a movement of air which is directed into its opening. The intensity of this movement is a maximum at the opening and diminishes swiftly with the distance from it. In any plane in this region of influence, curves may be drawn which represent constant values of the air velocity. A series of such curves representing a velocity distribution are commonly called lines of velocity potential or contours of equal velocity. If a set of curves be traced so as to be perpendicular at every point of intersection with these contours, they are designated as stream lines, or lines whose tangent at any point gives the direction of flow. Between two such curves the volume of flow is constant. Thus, in Fig. 1 indicating a diametrical plane in the region of influence of a duct end under suction, the curves *A, B, C,* etc., are equal velocity contours having the values *A, B, C,* etc., and the curves *0, 1, 2, 3,* indicate the direction of flow along a succession of points lying on them. Moreover, as has been stated, the flow across a section *pq* is equal to the flow across *mn, jk,* etc.

From the geometric symmetry of a circular opening, it is easily seen that the velocity conditions existing in Fig. 1 are the same through any other diametrical plane. Consequently, the rotation of the plane about the axis will generate surfaces of equal velocity and tubular sheets of flow. Thus, the velocity and stream line distributions in a single radial plane of a circular opening represents the conditions existing throughout the sphere of influence. Further, it can be shown [2] that whatever the size of the duct end, or whatever the volume of flow, the velocity contours and the stream lines are always of

---

[1] Department of Public Welfare, City of Cleveland, Ohio.

[2] Studies in the Design of Local Exhaust Hoods, by DallaValle and Hatch. (A paper presented at the Sixth Annual Wood-Industries Meeting, *American Society of Mechanical Engineers,* October 15, 16, 1931.)

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Milwaukee, Wis., June, 1932.

the same general form. These facts which have been experimentally determined, may be extended to all hoods and condensed into the following theorem: *The positions of the velocity contours for any hood when the contours are expressed in terms of the velocity at the hood opening are purely functions of the shape of the hood; the contours are identical for similar hood shapes when such hoods are reduced to the same basis of comparison.* For example, if the velocity 4 in. outward along the axis of an 8-in. diameter duct end is 26 per cent of the velocity in the plane of the opening, the same value must occur for a 12-in. diameter duct end at a distance of 6 in. In other words, if the

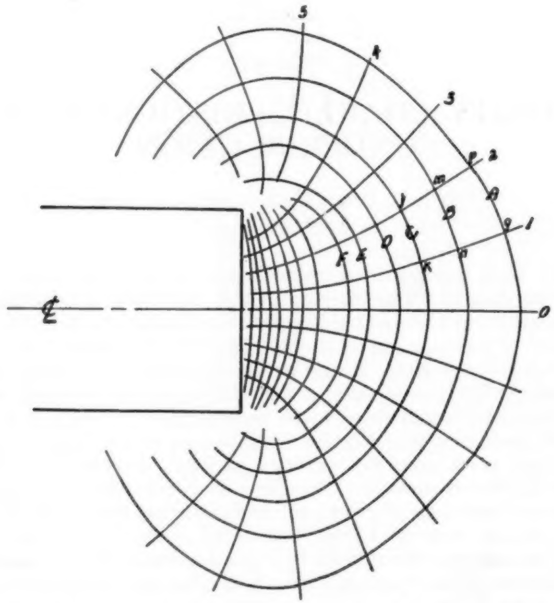


FIG. 1. VELOCITY CONTOURS AND STREAM LINES FOR A CIRCULAR DUCT END

diameters of each opening be divided into a stated number of parts and if each field of influence be divided in corresponding units, the contours will be identical.

An extension of these important facts to rectangular hoods appears difficult at first since velocity distributions will vary in every radial plane drawn through a whole quadrant. However, in corresponding planes, this must necessarily hold true. In such cases, it is convenient to consider the distributions in two center planes at right angles to each other. These planes are designated according to the side to which they are perpendicular. In square openings, of course, the velocity distributions in two such planes must be the same.

EFFECT OF SHAPE ON THE DISTRIBUTION OF VELOCITY CONTOURS

It may reasonably be supposed that the amount and the degree of flaring of a hood has considerable effect on the distribution of the velocity contours. If, however, it is assumed that the distributions of flow across two *openings* of the same size and shape are similar, the velocity contours are substantially the same, although the degree of flare in each differs. The main portions of the sphere of influence which are affected lie behind the edge of the opening, the boundary surface of the flared portion being the chief variable. The velocity contour distribution, therefore, may be said to be dependent upon the shape of the opening only, provided the distribution of flow across it does not vary greatly with the degree of flare. As a matter of experiment, it has been

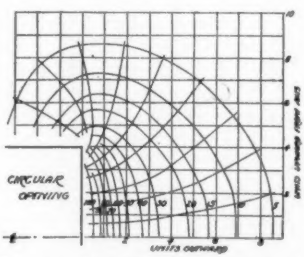


FIG. 2. VELOCITY CONTOURS AND STREAM LINES IN A RADIAL PLANE OF A CIRCULAR OPENING. CONTOURS ARE EXPRESSED AS PERCENTAGES OF THE AVERAGE VELOCITY IN THE PLANE OF THE OPENING

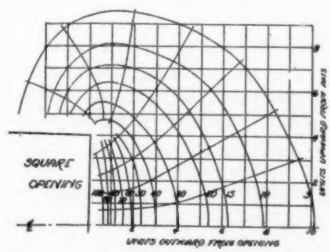


FIG. 3. VELOCITY CONTOURS AND STREAM LINES IN A RADIAL PLANE PERPENDICULAR TO ONE OF THE SIDES OF A SQUARE OPENING

found that over a very wide range the degree of flare alters the distribution of flow across an opening only slightly.[2] Hence, a 5-in. by 10-in. opening with a 12-in. flare to a 3-in. diameter duct gives the same contour distribution forward of an opening as one with an 8-in. flare (say) to a 5-in. diameter duct. In fact, were the opening but the end of a 5-in. by 10-in. duct, the velocity contour distribution would differ only slightly. Since, as a rule, most hoods are gradually flared in order to reduce entrance losses, it is practical when dealing with contour distributions, to express the overall shape of the hood in terms of the shape of its opening. In the case of rectangular openings, the shape will be referred to in terms of the ratio of the short side to the long side, a 5-in. by 10-in. opening, for example, having a ratio of sides equal to ½, and is so designated.

---

[2] Studies in the Design of Local Exhaust Hoods, by DallaValle and Hatch. (A paper presented at the Sixth Annual Wood-Industries Meeting, *American Society of Mechanical Engineers*, October 15, 16, 1931.)

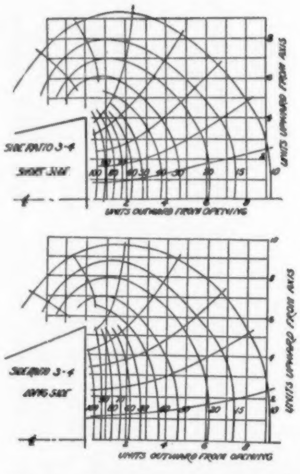


FIG. 4. VELOCITY CONTOURS AND STREAM LINES FOR A RECTANGULAR OPENING WHOSE RATIO OF SIDES IS 3 TO 4

b. (*above*) Contours and Stream Lines in Radial Plane Perpendicular to Short Side of Opening

a. (*below*) Contours and Stream Lines in Radial Plane Perpendicular to Long Side of Opening

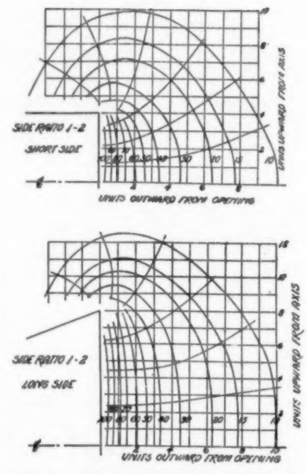


FIG. 5. VELOCITY CONTOURS AND STREAM LINES FOR A RECTANGULAR OPENING WHOSE RATIO OF SIDES IS 1 TO 2

b. (*above*) Contours and Stream Lines in Radial Plane Perpendicular to Short Side of Opening

a. (*below*) Contours and Stream Lines in Radial Plane Perpendicular to Long Side of Opening

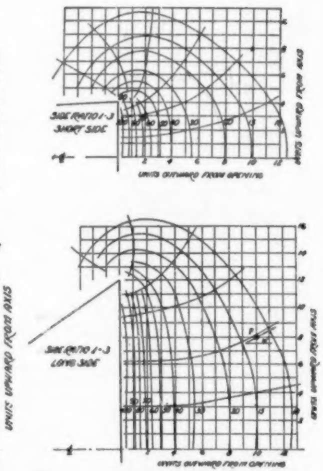


FIG. 6. VELOCITY CONTOURS AND STREAM LINES FOR A RECTANGULAR OPENING WHOSE RATIO OF SIDES IS 1 TO 3

b. (*above*) Contours and Stream Lines in Radial Plane Perpendicular to Short Side of Opening

a. (*below*) Contours and Stream Lines in Radial Plane Perpendicular to Long Side of Opening

DISCUSSION

The velocity contours and stream lines for various types of hood openings are shown in Figs. 2 to 7 in the manner already discussed. Thus, provided the region of influence for all round openings is mapped in the coordinate units corresponding to Fig. 2, the velocity contours will all be identical and will be expressed in per cent of the average velocity in the plane of the opening. The same is true for the various shaped rectangular hoods where the short side of each is divided into eight Units, and the long side in the same units in proportion to its length. An opening, therefore, with a ratio of sides of 3 to 4 will have $\frac{8 \times 4}{3} = 10\,2/3$ units on the long side.

An example will illustrate the application of the contour lines of Figs. 2 to 7. Suppose it is desired to find the speed and direction of the air movement at a point 4 in. upward and 5 in. outward in the radial plane perpendicular to the short side of a 4-in. by 12-in. hood handling 1,200 cfm. The ratio of sides is ⅓ and expressing the coordinates of the point on the basis that there are 8 units of length on the short side the coordinates of the point in question are 8 units upward of the axis and 10 units outward, as shown by the circle in Fig. 6b. The velocity at this point is approximately 12½ per cent of the velocity at the opening. Hence, the velocity at the opening being $\frac{1200 \times 144 \times 12}{4} = 3600$ fpm, the velocity at the point must be $0.125 \times 3600 = 450$ fpm. The direction of flow may be obtained by drawing the tangent to the stream line at this point and is found to be equal to 30 deg of arc. These calculations assume that the flow in the region of influence is not grossly impeded by an obstruction. Otherwise, the contours are altered and must be redetermined with the obstruction in place.[3] It may be said, however, that if the obstruction is considerably smaller in cross sectional area than the surface area of the contour passing through it, the velocity distributions as shown in the figures are not substantially altered. The farther the obstruction is removed from an opening, the less is the error involved.

A matter of considerable importance is the velocity distribution over the hood opening. For the purpose of simplifying calculations, it has been assumed in developing the contours in Figs. 2 to 7, that the velocity distribution over the opening was uniform. Such an assumption is not correct as may be judged from an examination of the figures themselves, which show the 100 per cent contour to be somewhat displaced from the opening. In other words, the figures show a higher velocity than the average over the central portions of the opening. The situation is somewhat like the phenomenon occurring when a fluid flows through a pipe, which because of viscosity gives a higher velocity at the axis than toward the edges. The effect in the case of hoods does not, however, arise from similar considerations. In hood openings, it is necessary to contend with an *edge* effect. Air entering from behind the hood is forced to turn abruptly into it, thus creating a stationary vortex which restricts the effective area to a value less than the actual. At the corners of a

[3] For an example of the type of obstruction included in this statement, see a paper by Hatch, Drinker and Choate, entitled, Control of the Silicosis Hazard in the Hard Rock Industries, I. Journal of Ind. Hygiene, XII, *3*, March, 1930, p. 87.

rectangular hood, the effect may be considered as intensified. Experimental data tend to show that the effective area is reduced in proportion to the perimeter of the opening.

The *edge* effect may be considerably reduced by the use of a flange placed

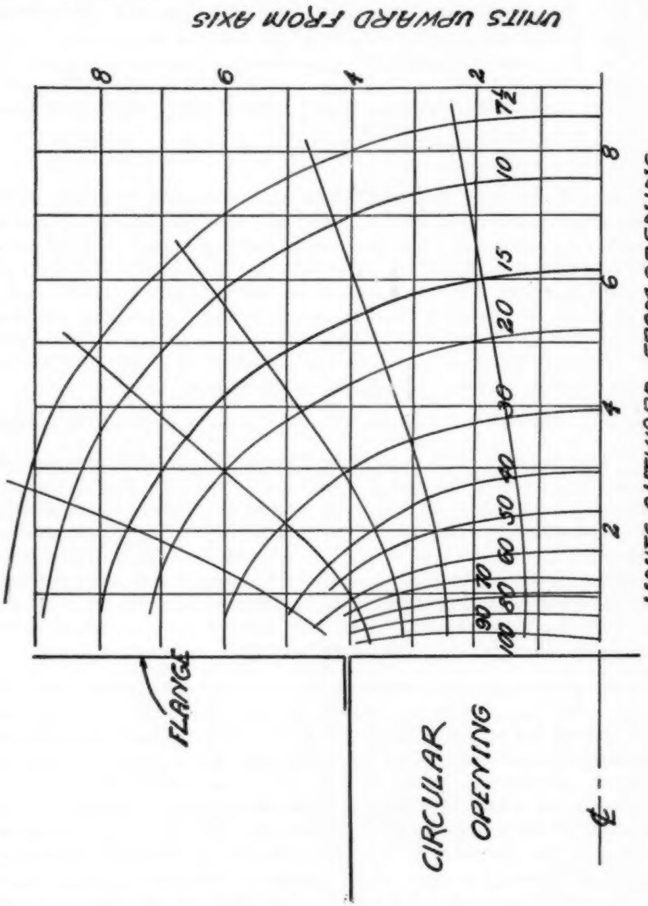


FIG. 7. VELOCITY CONTOURS AND STREAM LINES IN A RADIAL PLANE OF A CIRCULAR OPENING WITH A RIGHT ANGLED FLANGE. COMPARE WITH FIG. 2.

around the edge and lying in the plane of the opening. A flange approximately 5 in. in width is sufficient for hoods up to 3 sq ft in area. The flange tends to cut off the flow from the region behind the opening which is frequently useless, and is advantageous in two respects: *first,* it increases the effectiveness of the hood in the forward regions, and *second,* it reduces the energy consumption of the hood. In Fig. 7 the velocity contours and stream lines of a flanged

circular opening are shown, and may be contrasted for the sake of clearness with the characteristics for the same opening in Fig. 2.

FORMULA AND NOMOGRAPH TO DETERMINE THE AXIAL VELOCITIES OF HOODS

An examination of Figs. 2 to 7 shows the interesting fact, that over the region projected by the edges of the opening, the velocity contours are parallel

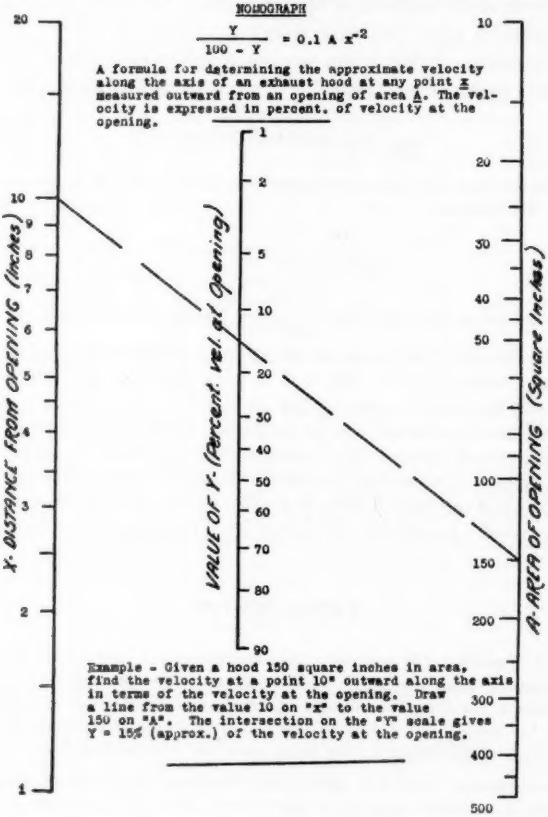


FIG. 8. NOMOGRAPH FOR THE CALCULATION OF Y WHEN x AND A ARE GIVEN IN THE FORMULA

to its plane. It can be seen, therefore, that a formula giving the velocity outward along the axis of a hood under suction would be of considerable value. Such a formula has been developed for rectangular and round openings.[2] For the former, the formula has been found to be,

[2] Studies in the Design of Local Exhaust Hoods, by DallaValle and Hatch. (A paper presented at the Sixth Annual Wood-Industries Meeting, *American Society of Mechanical Engineers,* October 15, 16, 1931.)

$$\frac{Y}{100-Y} = 0.0833\left[\frac{1}{1+0.259\left(\frac{r}{1-r}\right)^{-1.104}}\right]A^{1.04}\ x^{-1.91} \qquad (1)$$

where

x = the distance outward along the axis measured in inches

A = the area of the opening in square inches

r = the ratio of sides (less than unity)

Y = the per cent velocity at the opening found at the point x

For circular openings, expressing A in terms of the diameter d, the formula is,

$$\frac{Y}{100-Y} = 0.0645d^{2.10}\ x^{-1.91} \qquad (2)$$

A convenient and approximate formula, suitable for all shapes of openings is given by the relation

$$\frac{Y}{100-Y} = 0.1A\ x^{-2} \qquad (3)$$

In these formulas the function $\frac{Y}{100-Y}$ has been introduced in order that the physical limits of the problem might be satisfied. Thus, when $x = \infty$, $f(Y) = 0$, and when $f(Y) = 100$, $x = 0$. The fact that the velocity variation outward along the axis is very nearly as the inverse square of the distance, suggests that hood problems may be treated as though the openings were electrostatically charged plates. The mathematical treatment, however, except in very special cases, is exceedingly difficult. In order to simplify the calculation of Y, x or A when any two of these variables are known, the nomograph, Fig. 8, has been prepared. Its manner of use has been indicated in the figure.

DISCUSSION

PROF. A. I. BROWN[4] (WRITTEN): The mathematical analysis of the velocity at different distances from hoods of various shapes no doubt will find application in specific problems involving the design of hoods and their location with respect to the area to be vented, but I believe this paper is of particular interest in emphasizing the effect of the shape of the hood upon the direction of flow.

The stream lines of Figs. 1-7 extend only to the plane of the opening but their direction clearly indicates what takes place within the hood or duct. In Fig. 2 the contraction of the stream which is caused by the abrupt entrance is clearly shown by the stream lines near the edge of the opening. The decrease in the size of the stream or the increase in the entrance loss due to an abrupt change in direction is well known and yet too often overlooked.

An increasing observance of the importance of the application of aerodynamics principles to sheet metal duct work is apparent and is encouraging. The author of this paper is to be commended for presenting an accurate analysis of what takes place in the vicinity of hoods of various shapes.

[4] Associate Professor of Heating and Ventilating, Ohio State University, Columbus, O.

L. T. M. Ralston (Written): This is the first time in the writer's experience that such detailed and informative data on this subject have been prepared and the entire paper is a distinct contribution to the ventilating art and one which should be extremely useful to engineers in designing proper type of exhaust openings for exhaust hoods over equipment such as urns, ranges, laboratory tables, etc., and in conjunction with industrial and school exhaust systems.

The data in this paper can be used with profit by certain manufacturers of hoods so as to insure proper design for the most efficient means of collecting air and easy flow to the duct connections. The charts and formula are easy to use and adequately fulfill the need which has heretofore been unfilled in an interesting detail of ventilation.

C. A. Booth (Written): The information contained in this paper is very useful, and the results will be instructive to those who are not familiar with air flow but may have occasion to design or use exhaust systems requiring a minimum air velocity distributed over a definite area. The cases treated are all open end pipes without obstructions, and it would add materially to the value of this paper if the diagrams included cases where obstructions in the hood openings parallel some of the typical exhaust hood installations, such as emery and polishing wheels, pots, and tanks.

President Rowley: The paper presents data which are of value for effective hood design. Any one who has attempted this work or has observed the results obtained, realizes that some such fundamental data should be available.

J. J. Robson: Do you use an ordinary Pitot tube to determine the velocity at the points indicated?

Mr. DallaValle: The ordinary Pitot tube, specified by the Standard Fan Code, would not measure velocity at the point. I used a bi-sectional cylindrical tube which was merely a tube ⅛ in. in diameter and about 2 ft long, partitioned through its whole length into two independent parts. At the mid-section, two holes, about two-hundredths of an inch in diameter, were drilled opposite each other, one for each section of the tube. The tube was placed with the holes parallel to the direction of the flow, and gave readings almost twice as great as the ordinary standard. The factor for the tube used was 1.75 in contrast with 1.0 for the Standard Tube. The tube was sufficiently small and could be placed at a point in space for velocity measurements. (See *A.S.M.E. Transactions.*) The tube has been discussed very much abroad, but very little has been done with it in this country.

Mr. Robson: It is not a new piece of apparatus that you designed yourself?

Mr. DallaValle: No, but there is a modification of it that we were responsible for designing.

C. J. Fechheimer: I would like to ask the author in connection with the tube that he used, whether he integrated velocities graphically or some other way, averaged up the velocities and compared the average times the area with the volumes measured by some other means?

Mr. DallaValle: I didn't do that. It never occurred to me as a matter of fact. I might add that I took two square openings and put them edge to edge which would give an opening with the ratio of sides of one to two and combined those two square openings graphically, knowing their initial velocity contour distribution, and I was able to reproduce the same contour distribution as I got with the one to two as shown in Fig. 5b.

Mr. Fechheimer: My next question is, what kind of manometer did you use for the very small velocities?

Mr. DallaValle: I used a Whalen gage.

MR. FECHHEIMER: Did you have reason to believe that the flow in most cases was laminar or turbulent?

MR. DALLAVALLE: I think it is laminar, but as you get close to the opening the stream lines maintain their uniformity on account of the stiffness factor. The flow near the opening is, however, turbulent.

No. 932

NATURAL WIND VELOCITY GRADIENTS NEAR A WALL

By J. L. Blackshaw [1] and F. C. Houghten,[2] Pittsburgh, Pa.

MEMBERS

IN 1931 the Research Laboratory of the American Society of Heating and Ventilating Engineers, in co-operation with the United States Bureau of Mines, published a paper [3] which included a study of wind velocity gradients resulting from the frictional effect of the wind near a wall in free space. The results of this investigation were useful in converting film resistance coefficients obtained with apparatus having confining ducts to coefficients applicable to open air conditions found on actual building surfaces.

In setting up the apparatus for the earlier study, it seemed impractical to arrange the equipment on the wall of a building and wait for favorable winds with which to determine velocities; a wind tunnel was not considered because it would produce the undesired duct condition. Rather than to move air past a test wall, it was decided to move a test panel through the air and thus artificially produce an effect similar to that of natural wind blowing parallel to a wall. This was done by driving a truck, with a panel erected on its side, at uniform velocities along a level stretch of road at times when there was no natural wind blowing. This earlier paper contained, in addition to information on velocity gradients obtained with the truck, other data covering velocity gradients in ducts. Velocity gradient curves obtained in the earlier study are shown in Fig. 1 for a surface in free space, in a 12-in. duct, and in a 6-in. duct for wind velocities of 10, 20, and 30 mph, respectively.

Some question was raised concerning the application of the results obtained by the survey made with the moving truck to conditions of natural air flow near an actual wall. There was the possibility that waves of air set up by the front of the truck and eddy currents from its undercarriage would have affected the data. With this in mind, the Committee on Research and the

[1] Research Engineer, A.S.H.V.E. Laboratory, Pittsburgh, Pa.
[2] Director, A.S.H.V.E. Laboratory, Pittsburgh, Pa.
[3] Wind Velocity Gradients Near a Surface and Their Effect on Film Conductance, by F. C. Houghten and Paul McDermott. (A.S.H.V.E. Transactions, Vol. 37, 1931).

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Milwaukee, Wis., June, 1932, by J. L. Blackshaw.

Technical Advisory Committee on Heat Transmission authorized a new study on velocity gradients in free space to be made under natural wind conditions using a stationary panel located at a vantage point subjected to free air flow.

A search was made about Pittsburgh and its environs to find a suitable spot for the erection of a swiveled test panel where it would be exposed to unrestricted natural winds. The bald top of one of Pittsburgh's hills could have

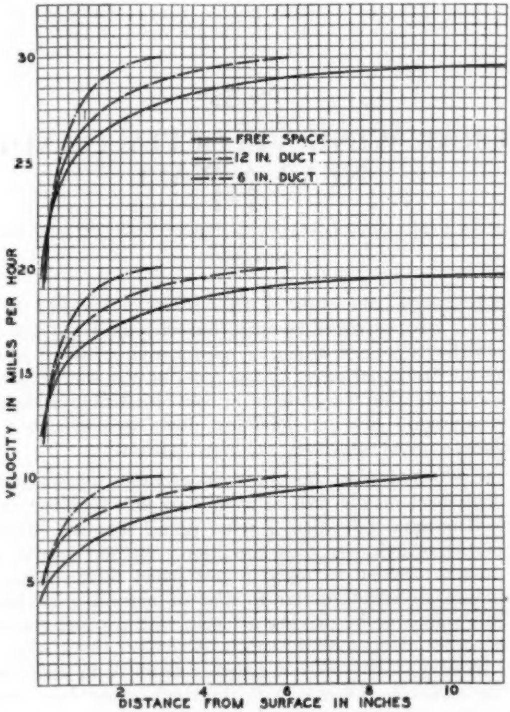


FIG. 1. VELOCITY GRADIENTS FOR A SURFACE IN FREE SPACE AND IN 12-IN. AND 6-IN. DUCTS

been used, but the roof of an isolated high building was considered more adaptable. Through the courtesy of Chancellor Bowman of the University of Pittsburgh, the top of the Cathedral of Learning of the University was made available to the Laboratory for this project. The Cathedral of Learning, now being constructed at a cost of more than $8,000,000 has 41 stories and is 535 ft high. (See Fig. 2).

At the top of this building is a 30 ft square observation platform that is entirely clear of all obstructions except a small temporary elevator scaffold and an airplane beacon, neither of which blankets more than a few feet of

wind flow area. This platform offered an ideal location for the test equipment, which was similar to that built on the moving truck in the earlier study as to the type of panel and velocity tubes used and the photographic method employed to simultaneously record the pressures indicated by the velocity measuring tubes.

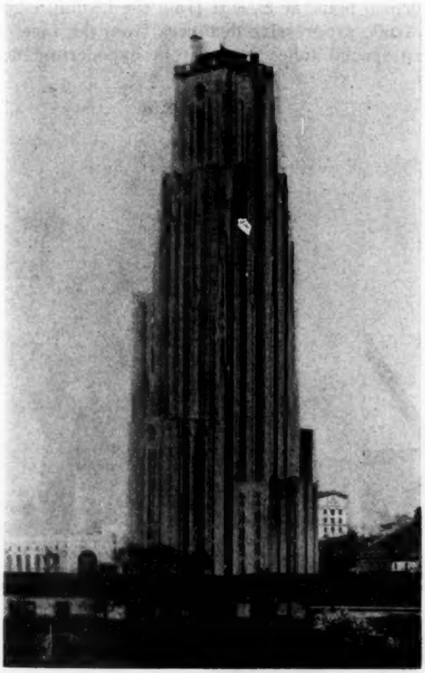


FIG. 2. THE CATHEDRAL OF LEARNING OF THE UNIVERSITY OF PITTSBURGH PHOTOGRAPHED FROM THE OFFICE WINDOW OF THE A. S. H. V. E. RESEARCH LABORATORY LOCATED IN THE PITTSBURGH EXPERIMENT STATION OF THE U. S. BUREAU OF MINES

DESCRIPTION OF APPARATUS

The front of the test panel in position for this work is pictured in Fig. 3, and a line drawing of the set-up is shown in Fig. 4. The rigid ¼-in. thick composition board panel, *A*, 6 ft high and 12 ft long, painted with gray wall paint, was held securely 2 ft above the floor by light, braced framework. This framework, designed to withstand a 90-mph wind blowing perpendicular to the panel, was firmly anchored to the concrete floor of the observation platform at *B* with a swivel coupling, which permitted the panel and recording

apparatus to be turned through a complete revolution to bring the face of the panel parallel to the direction of air flow. A wind vane, *C*, indicated the direction of the air stream by means of a pointer. The cup anemometer, *D*, recorded the average wind velocity during any test run.

For the study on velocity gradients 12 velocity measuring tubes were held in the panel in a vertical plane at *E*, 8 ft from the leading edge of the panel, *F*, and were set to protrude progressive distances from the face of the panel. Six of these tubes were special tubes made from hypodermic needle tubing with

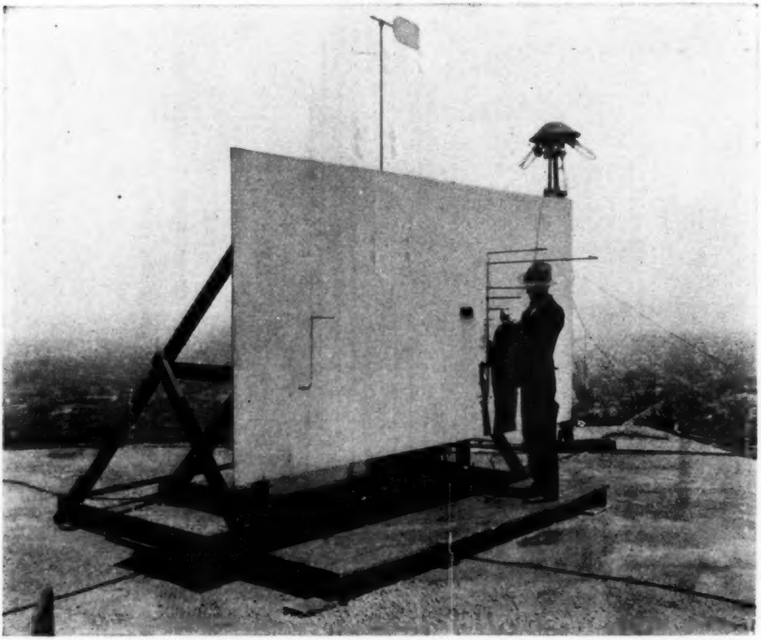


FIG. 3. FRONT OF TEST PANEL USED IN STUDY OF VELOCITY GRADIENTS

an inside diameter of 0.0531 in. and an outside diameter of 0.0720 in., which, having so little bulk, allowed pressures to be measured close to the wall with little interference of the air stream. They recorded total pressures only. Static pressures were determined in a static pressure box, *G*, which had fourteen 0.040 in. holes drilled in it perpendicular to its ⅛-in. brass face, which was sunk flush into the panel near the tubes. If this static chamber, *G*, were connected to one leg of an inclined draft gage, and a total pressure tube were connected to the other, the displacement of the liquid in the gage would give the velocity pressure from which the velocity of the air could be determined. The other 6 tubes were standard Pitot tubes having an outer tube to indicate static pressure and an inner one to indicate total pressure. If the outer

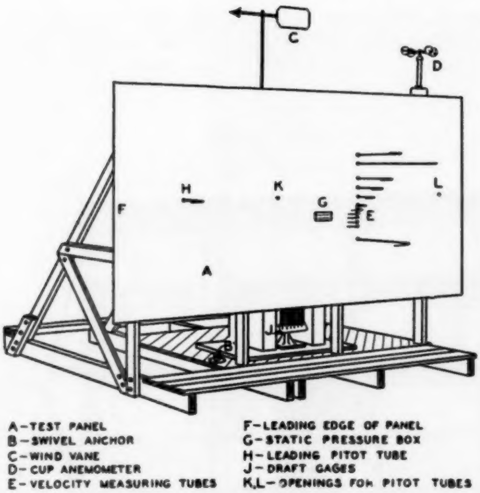


FIG. 4. SET-UP USED IN STUDY OF VELOCITY GRADIENTS

tube were connected to one leg of an inclined draft gage and the inner tube were connected to the other, the displacement of the liquid in the gage would give the velocity pressure from which the velocity of the air could be determined. A thirteenth standard Pitot tube, *H*, was held at a point half-way up the panel but only 2 ft from the leading edge. In the set-up for velocity gradients, the type of tube and its position, protrusion, and distance from the floor and the leading edge are given in Table 1.

TABLE 1

Tube No.	Kind of Tube	Protrusion (Inches)	Distance from Floor	Distance from Leading Edge (Feet)
1	Standard Pitot	20	3 ft. 10½ in.	8
2	Special	⅛	4 ft. 2½ in.	8
3	Special	¼	4 ft. 4 in.	8
4	Special	½	4 ft. 5½ in.	8
5	Special	1	4 ft. 7½ in.	8
6	Special	2	4 ft. 8½ in.	8
7	Special	4	4 ft. 10 in.	8
8	Standard Pitot	8	5 ft. 1 in.	8
9	Standard Pitot	12	5 ft. 4 in.	8
10	Standard Pitot	16	5 ft. 7 in.	8
11	Standard Pitot	36	6 ft. 0 in.	8
12	Standard Pitot	24	6 ft. 3 in.	8
13	Standard Pitot	8	5 ft. 1 in.	2

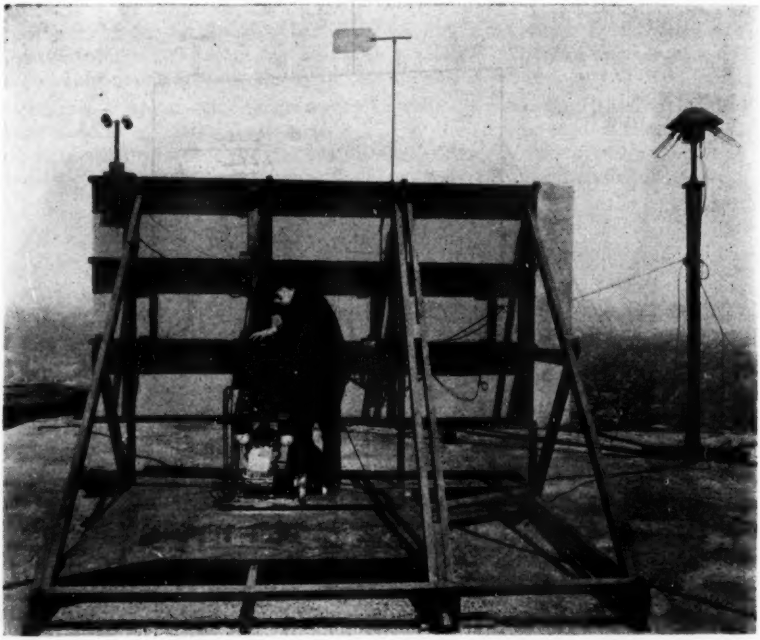


FIG. 5. BACK OF TEST PANEL USED IN STUDY OF VELOCITY GRADIENTS

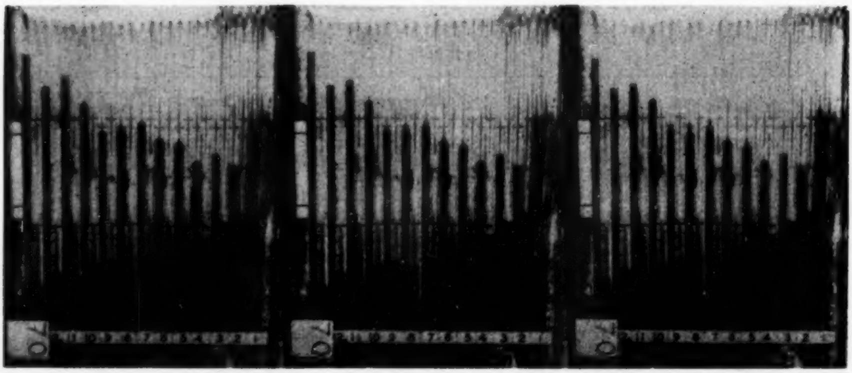


FIG. 6. MOTION PICTURE FILM SHOWING TYPICAL MANOMETER GAGE DISPLACEMENTS

In Fig. 4, *J* indicates an adjustable inclined manometer frame which supported 13 draft gages to which all 13 velocity tubes were connected by rubber tubing. Obviously it would be impossible for an observer to read all 13 gages at the same time, especially in a varying wind, so they were read photographically by using a light bank and a motion picture camera in a manner similar to that used in the previous study. This method again proved entirely satisfactory. Fig. 5 is a photograph of the back of the test panel showing the connections and apparatus used.

A short strip of film obtained by the photographic method is shown in Fig. 6. After development, each picture was enlarged by projection onto a screen where the displacement of the colored alcohol in the gages could be accurately read with the aid of a calibrated scale. The two lines at the left of each individual picture of the film strip are gage points for setting the scale in taking displacement readings. The numbers at the lower left hand corners refer to data recorded at the time the picture was taken, covering wet- and dry-bulb temperatures, the angle at which the manometer was set, the barometric reading, or any other observations made. Data were disregarded whenever successive pictures showed that the wind velocity or direction had become too variable for test purposes.

The velocity of the air was determined through the use of the formula:

$$V = 12.4568\sqrt{\frac{h \times \sin \alpha \times D}{w}} \qquad (1)$$

where

V = velocity of the air in miles per hour.

h = displacement of the gage fluid in inches.

α = angle the manometer made with the horizontal.

D = specific gravity of the gage fluid at the temperature of the gage.

w = weight of the air in pounds per cubic foot for the prevailing wet- and dry-bulb temperature and barometric pressure.

In formula 1, w may be quickly determined without a laborious mathematical process by reference to the Younger chart.[4]

TEST PROCEDURE

In taking data, the panel was turned until the wind vane indicated a parallel flow of air past the tubes. All tubing and connections were checked for possible leaks. After leveling the inclined manometer and recording the angle at which it was set, the operator waited until a parallel and steady wind gave a fairly constant displacement of the fluid in the gages. In photographing the manometer, the camera was adjusted to record approximately 2 pictures a second.

In a preliminary run to calibrate the tubes, all 13 velocity tubes were set so they protruded an equal distance from the panel, and data were taken from which a calibration curve could be drawn for each tube to correct it

[4] Chart for Determining the Weight of Moist Air, by John E. Younger. (*Mechanical Engineering*, June, 1925, p. 492.)

for position and to even out any slight differences which might be found among the tubes. In a second preliminary run, tubes No. 9 and 10 were taken out of the vertical plane, *E*, and again protruded equidistant from the panel at points *K* and *L* in a horizontal plane midway of the panel, in order that readings of the gages would show any possible horizontal wind velocity gradient. In the test none was found.

After this preliminary information had been obtained, the tubes were adjusted in the panel to protrude the respective distances given in Column 3 of Table 1. Pitot Tube 13 protruded the same distance from the panel as Pitot Tube 8 and was in the same horizontal plane with it; therefore, identical readings of tubes 8 and 13 indicated an air flow parallel to the panel. Because Pitot Tube 1, lowest in the line of tubes, stuck out nearly as far as the two upper tubes, it recorded any disturbing effect of turbulence or diagonal air flow across the panel.

The solid line curve in Fig. 7 connects 12 points showing the respective positions of the 12 tubes and the velocities which were indicated by them experimentally in a single test. The curve gives a typical velocity gradient away from the panel for a free wind velocity of 20.8 mph.

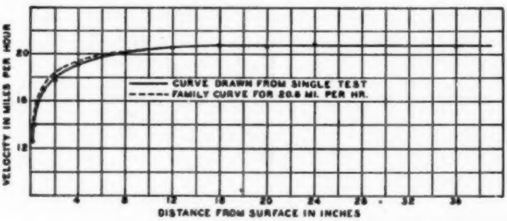


FIG. 7. WIND VELOCITY GRADIENT FOR SINGLE TEST AND FAMILY CURVE FOR THIS VELOCITY

DATA AND RESULTS

Fig. 8 gives typical experimental data showing the relation between the velocity in free space and the velocities found at points ⅛-in. and 12-in. from the panel, as determined respectively by Tubes No. 2 and No. 9 in a series of test runs. The free wind velocities were taken as the velocities given by Tube No. 11. Tubes 1, 10 and 12 indicated the same velocities as Tube 11 which showed that the air stream beyond Tube 10 was unaffected by the resistance of the surface.

Curves similarly plotted are shown in Fig. 9 for all 12 tubes to compare free wind velocities with velocities at the varying distances from the surface given as *X* in the drawing. The curves tend to pack together as the distance from the surface increases. Except at the higher velocities, these curves check almost exactly with similar curves obtained from data taken with the moving truck in the earlier work. The fanning of the curves at the higher velocities might be due to the greater air speed obtained with the artificially produced wind created by the truck.

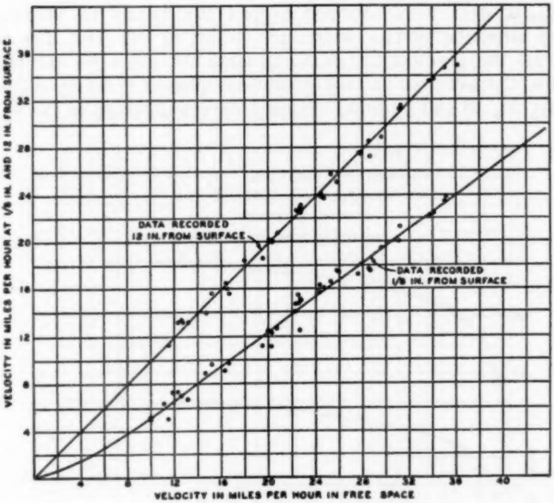


FIG. 8. RELATION BETWEEN VELOCITY IN FREE SPACE AND VELOCITIES AT POINTS 1/8-IN. AND 12-IN. FROM THE SURFACE

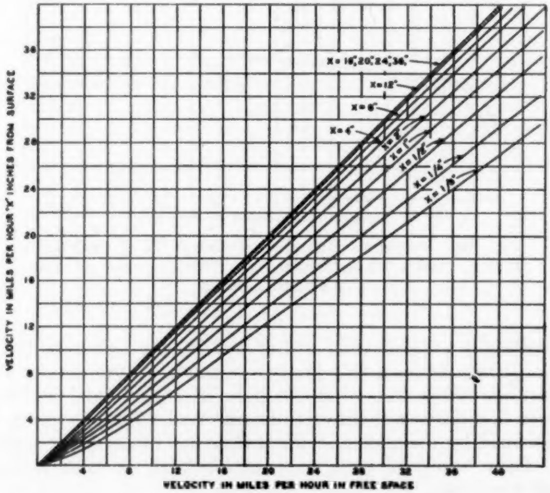


FIG. 9. RELATION BETWEEN VELOCITY IN FREE SPACE AND VELOCITIES AT POINTS AT WHICH VELOCITY TUBES WERE LOCATED

From Fig. 9, data were obtained to plot the family of velocity gradient curves away from the surface which are shown in Fig. 10. The heavy solid line curves are the gradient curves away from the panel for the natural wind on top of the Cathedral of Learning, and are plotted for free wind flow conditions of 10, 20, 30 and 40 mph. Similar curves obtained in the earlier work

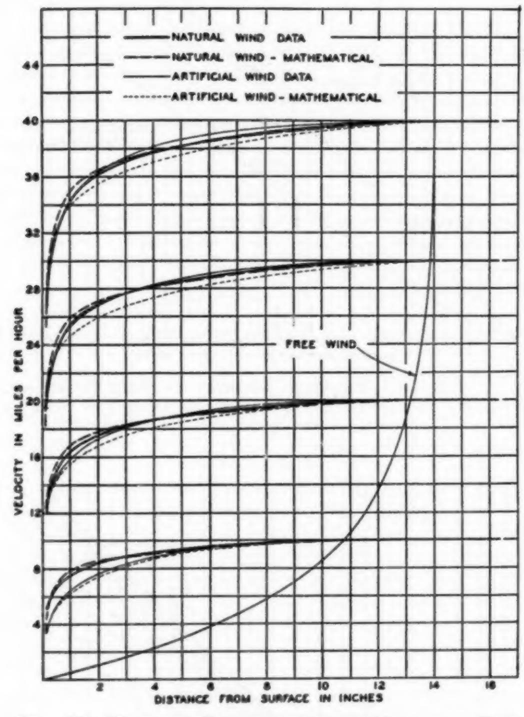


FIG. 10. VELOCITY GRADIENT CURVES NEAR THE TEST PANEL SHOWING COMPARATIVE FAMILY AND MATHEMATICAL CURVES

with the artificially created wind are drawn in this figure as the light solid line curves. It may be seen that the two separate studies check closely except at low velocities. At a free wind condition of 10 mph and at 1-in. from the surface, there is a difference between the curve of 1.5 mph, which discrepancy may be explained by the fact that the measuring tubes are insensitive at low velocities. Data below 12 mph were extrapolated in both the natural and the artificial wind tests.

At any desired velocity in free wind, a family velocity gradient curve may be read from Fig. 9. Such family curve was read for 20.8 mph free wind

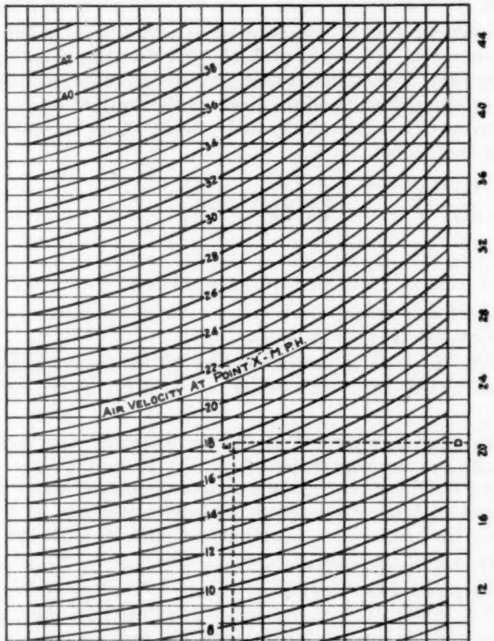


FIG. 11. CHART FOR DETERMINING AIR VELOCITY GRADIENTS IN DUCTS AND IN FREE SPACE

velocity and was plotted in Fig. 7 together with the typical experimental curve at 20.8 mph drawn from actual test data. The curves lie closely together.

As the data shown in Fig. 9 consisted of straight lines passing through different Y axis intercepts, it seemed probable that the data should fit a logarithmic curve. By using logarithmic paper and, in each case, plotting distance from the surface as one ordinate and the slope of the straight line as the other, a relationship between free wind velocity and velocity at any distance from the surface was determined to be:

$$V_x = V_f \left(\frac{X - 0.123}{13.877}\right)^{0.03623} - \left(\frac{1.4}{\sqrt{X}} - 0.27\right) \tag{2}$$

where

V_x = velocity of wind at X distance from the surface.

V_f = velocity of wind in free space.

X = distance from the surface in inches.

The accuracy of this formula is shown on Fig. 10 by the coincidence of the heavy dash line calculated curve with the heavy solid line curve derived from test data.

The mathematical curves presented in the earlier paper for the respective four free wind velocities are superimposed on Fig. 10 by the light dash lines. The curve marked *Free Wind* on Fig. 10 connects the points where the various velocity gradient curves become asymptotic and marks the end of the gradient effect caused by the surface, and the beginning of unrestricted wind flow.

A chart, Fig. 11, for correlating air velocity gradients in ducts and in free space has been developed from laboratory data. With this chart, any three of four factors (*i.e.*, width of duct, air velocity at center of duct, distance X from duct surface, and air velocity at distance X) may be combined to give the fourth factor. Since it has been shown that beyond 14 in. the resistance offered by a surface has little appreciable effect on the air stream, a duct 28 in. wide or larger should give a gradient condition similar to that of free space. On the chart, the free space line is identical to that for the 28 in. duct.

Example: What is the air velocity 1 in. from the surface of a duct 12 in. wide, when the air velocity at the center of the duct is 20.5 mph?

Solution from dotted line on chart: Find width of duct, 12 in. at A. Follow ordinate to intersection with distance from surface, 1 in. at B. Follow oblique line to intersection with heavy line at C. Find E, 17.6 mph, or the required velocity 1 in. from the surface, by the intersection of the abscissa CE and the ordinate DE which is drawn through the velocity at the center of the duct, 20.5 mph.

CONCLUSIONS

The study on velocity gradients made under natural wind flow conditions on top of a high building shows substantially the same results as those obtained under artificial wind flow conditions produced with a moving truck. A smooth plane wall section was used in both set-ups, and it is possible that the gradient curves obtained would have to be adjusted when figuring for rough walls. A formula for free space and a chart for ducts and free space

are given to show the relation between free wind velocity and velocity at any distance from the surface.

ACKNOWLEDGMENTS

The authors acknowledge the co-operation extended by Prof. J. A. Dent, John Weber, and J. E. McLean of the University of Pittsburgh in arranging for the study at the Cathedral of Learning.

DISCUSSION

PRESIDENT ROWLEY: This paper, on a subject which has been before us several times, is now open for discussion.

C. GEORGE SEGELER (WRITTEN): The data obtained in these and the previous experiments could be usefully correlated with heat transfer experiments both by natural convection and forced convection. It should also be possible to establish information with the new test panel on the building up of pressure against flat panels when directly in line with the wind.

In view of the difficulty of obtaining measurements accurately at points extremely close to the surface, it would be interesting to know whether the authors had explored the section up to ¼ in. distance from the wall more than indicated in the reported figures at ⅛ in. and ¼ in., especially due to the fact that at 0.1 in. from the surface Fig. 11 indicates that the width of the duct has no influence on the velocities closer to the surface than 0.1 in.

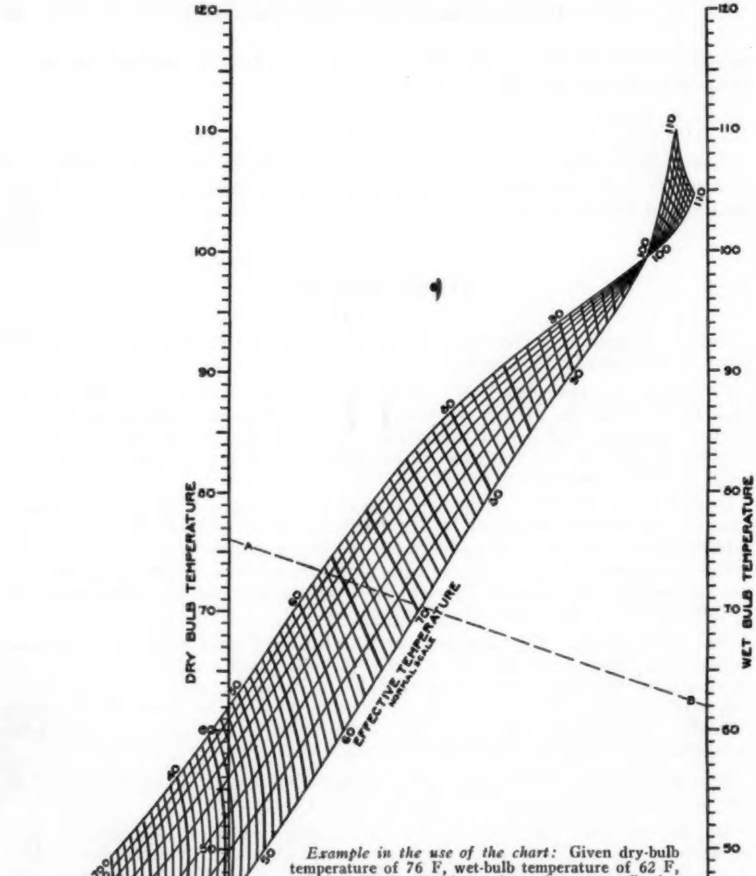
MR. BLACKSHAW: We could not obtain satisfactory data with our hypodermic needle Pitot tubes closer than ⅛ in. from the test panel. At this distance it was necessary to align these small tubes carefully to insure any sort of accurate result, and even then the erratic pulsating air flow so close to the panel made reading difficult. The curve in Fig. 11 is the extrapolation of an average of data taken at a greater distance with standard Pitot tubes which gave accurate readings.

WARREN EWALD: Did the area of that test panel affect your result? Would you get the same result from a small test panel as with a large one?

MR. BLACKSHAW: We have no way of checking results of different size panels as we built only the one 6 x 12 stationary test panel, and the results obtained with it checked with those previously had with the 5 x 9 panel on the moving truck. The tests on the larger panel were more conclusive. At one time it was hoped to make these studies on the side of a building, but it was decided that would be impossible because winds do not blow evenly on the sides of a building and the Pitot tube is useful only when wind is blowing parallel with it.

MR. EWALD: I would think offhand that a larger panel would give a different result than a small panel.

MR. BLACKSHAW: We talked that matter over and decided that a 6 x 12 panel was sufficiently large for the purpose, but we did not check the matter because we lacked the means. It is quite an expensive thing to build a large panel.



Example in the use of the chart: Given dry-bulb temperature of 76 F, wet-bulb temperature of 62 F, velocity of air 100 fpm; determine: (1) effective temperature of the condition, (2) effective temperature with *still* air, (3) cooling produced by the movement of the air, (4) velocity necessary to reduce the condition to 66 deg effective temperature.

Solution: (1) Draw line *A-B* through given dry- and wet-bulb temperatures. Its intersection with the 100-fpm curve gives 69 deg for the effective temperature of the condition. (2) Follow line *A-B* to the right to its intersection with the 20-fpm velocity line, and read 70.4 deg for the effective temperature for this velocity or so-called still air. (3) The cooling produced by the movement of the air is 70.4 — 69.0 = 1.4 deg effective temperature. (4) Follow line *A-B* to the left until it crosses the 66-deg ET line. Interpolate velocity value of 340 fpm to which the movement of the air must be increased for maximum comfort.

Fig. 1. Thermometric Chart Showing Normal Scale of Effective Temperature. Applicable to Inhabitants of the United States Under Following Conditions:

A. Clothing: customary indoor clothing
B. Activity: sedentary or light muscular work
C. Heating methods: convection type, i.e. warm air, direct steam or hot water radiators, plenum systems.

No. 933

HOW TO USE THE EFFECTIVE TEMPERATURE INDEX AND COMFORT CHARTS

Report on the True Significance and Limitations of the Effective Temperature Index and Comfort Charts prepared by the Technical Advisory Committee on Re-Study of Comfort Chart and Comfort Line, by C. P. Yaglou, Chairman, W. H. Carrier, Dr. E. V. Hill, F. C. Houghten and J. H. Walker

SINCE the effective temperature indices and comfort zones were determined 7 years ago, there has been some confusion concerning the true significance, application and limitations of these findings, in the minds of those who have not followed the literature closely. This is due probably to the great number of progress reports on the subject and to insufficient qualification of the results and conclusions. The purpose of this report is to clarify the obscure points and to fix, in the light of present knowledge, desirable comfort standards for practical use.

Significance and Application of Effective Temperature Index

Effective temperature is an index of warmth or cold. It is not in itself an index of comfort, as it is often assumed to be, nor are the effective temperature lines necessarily lines of equal comfort. This is true because in determining this index, the subjects compared not the relative comfort, but rather the relative warmth or cold of various air conditions. Moist air at a comparatively low temperature, and dry air at a higher temperature may both feel as warm as air of an intermediate temperature and humidity, but the *comfort* experienced in the three air conditions would be quite different, although the effective temperature is the same. The intermediate condition may be entirely comfortable, but the other two would not measure up to the same standard.

Under extreme humidity conditions there seems to be a difference between sensations of absolute comfort and proper degree of warmth. In other words, human beings are not necessarily comfortable when the air is neither *too warm* nor *too cold.* Air of proper warmth may, for instance, contain excessive water vapor, and in this way interfere with the normal physiologic loss of moisture from the skin, leading to damp skin and clothing and producing more or less discomfort; or the air may be excessively dry, producing appreciable discom-

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Milwaukee, Wis., June, 1932, by F. C. Houghten.

fort to the mucous membrane of the nose and to the skin which dries up and becomes chapped from too rapid loss of moisture. According to the comfort experiments both at the A. S. H. V. E. Laboratory in Pittsburgh, and at the Harvard School of Public Health in Boston, effective temperature appears to be a fair index of comfort also, only within a humidity range of 30 to 60 per cent, approximately.

Briefly, *effective temperature* may be defined as an arbitrary index of the degree of warmth or cold felt by the human body in response to temperature, humidity, and movement of the air. Effective temperature is not a temperature at all; it is a composite index which combines the readings of temperature, humidity and air motion in a single value. It has been called *effective* temperature because experiments showed that it is this composite index which initiates the physiologic *effects* produced in the body by heat or cold, regardless of the temperature, humidity and air movement components. The numerical value of the effective temperature index for any given air condition is fixed by the temperature of saturated air which, at a velocity or turbulence of 15 to 25 fpm, induces a sensation of warmth or cold like that of the given condition. Thus, an air condition has an effective temperature of 65 deg when it induces a sensation of warmth like that experienced in practically still air of 65 F saturated with moisture.

In all reports of the A. S. H. V. E. Research Laboratory, the term *still air* signifies the minimum air movement that was possible to obtain in the Laboratory's psychrometric chamber. Actually, the air motion was between 15 and 25 fpm in all experiments, without qualification as measured by the Kata thermometer. This was not a linear movement of air but it represented the turbulence or eddy currents produced by the air change. Even in tightly sealed rooms, the natural air movement is not likely to fall below 10 fpm so long as there is a temperature or pressure difference between the air inside and outside the room. In order to avoid misunderstanding, the committee recommends that in all future papers and charts, the term *still air* be replaced by the actual air movement or turbulence as determined by the Kata thermometer.

Given the dry- and wet-bulb temperature and the velocity of air motion, the effective temperature may be determined from the thermometric chart (Fig. 1) or from psychrometric charts with the effective temperature lines for various air velocities superimposed.[1-2] All these charts should be used specifically as effective temperature charts and not as comfort charts. Those showing the comfort zones for air velocities other than 15 to 25 fpm (often referred to as still air) should be redrawn, omitting the comfort zones for reasons explained later.

The thermometric chart (Fig. 1) applies to average normal and healthy persons adapted to American living and working conditions. Application is limited to sedentary or light muscular activity, and to rooms heated by the usual American convection methods (warm air, plenum and direct hot water and steam heating systems) in which the difference between the air and wall surface temperatures may not be great. The chart does not apply to rooms

[1] Effective Temperature with Clothing, by C. P. Yaglou and W. E. Miller (A. S. H. V. E. TRANSACTIONS, Vol. 31, 1925, p. 89).
[2] A. S. H. V. E. GUIDE 1932.

heated by radiant methods such as the British panel system, open coal fires, and the like. It will probably not apply with adequate accuracy to races other than the white or perhaps to inhabitants of other countries where the living conditions, climate, heating methods, clothing, etc., are materially different from those of the subjects employed in experiments at the A. S. H. V. E. Research Laboratory at Pittsburgh.

The effective temperature index for persons doing medium or heavy muscular work, in still air, is given in another paper.[3]

Limits of Comfort Zone, Comfort Line and Comfortable Humidities

According to the definition of the A. S. H. V. E. Research Laboratory, the *extreme comfort zone* includes air conditions in which one or more of the experimental subjects were comfortable. The *average comfort zone* includes those air conditions in which the majority of the subjects (50 per cent or more) were comfortable. That particular effective temperature at which the maximum number of subjects was comfortable was called the *comfort line.*

In some publications the comfort line for the winter season is placed at 64 deg ET, while in others it is put at 66 deg ET. Similarly the lower boundary of the average comfort zone is put at 61 deg and 63 deg ET, and the upper boundary at 69 deg and 71 deg ET.

A reference to the original reports will show that the A. S. H. V. E. Research Laboratory has developed two different scales of effective temperature: the basic,[4-5-6] and the normal,[1] and but one *comfort zone.*[7] In determining the basic scale, the subjects were stripped to the waist to eliminate the complicated influence of clothing. In the normal scale, the subjects wore customary indoor clothing. This is the scale used in ordinary ventilation work. The basic scale is applicable to hot industries where men usually strip to the waist.

The winter comfort zone was determined in 1923 at the A. S. H. V. E., Research Laboratory from votes of men and women subjects, wearing customary indoor clothing. This was about 1½ years before the normal effective temperature scale was developed. For this reason, the comfort zone for persons normally clothed was temporarily superimposed on the basic effective temperature chart for persons stripped to the waist. On this basic scale, the limits of the average comfort zone were placed tentatively at 61 and 69 deg and the comfort line at 64 deg basic effective temperature.

In 1924 when the normal effective temperature scale was established, the laboratory recomputed the comfort zone data in terms of this new scale, and superimposed the zone on the new effective temperature chart where it belongs. According to this revision, the average winter comfort zone has been perma-

[3] Effective Temperature for Persons Lightly Clothed and Working in Still Air, by F. C. Houghten, W. W. Teague and W. E. Miller (A. S. H. V. E. Transactions, Vol. 32, 1926, p. 315).

[4] Determining Lines of Equal Comfort (Warmth), by F. C. Houghten and C. P. Yaglou (A. S. H. V. E. Transactions, Vol. 29, 1923, p. 163).

[5] Cooling Effect on Human Beings Produced by Various Air Velocities, by F. C. Houghten and C. P. Yaglou (A. S. H. V. E. Transactions, Vol. 30, 1924, p. 193).

[6] Effective Temperature Applied to Industrial Ventilation Problems, by C. P. Yaglou and W. E. Miller (A. S. H. V. E. Transactions, Vol. 30, 1924, p. 339).

[7] Determination of the Comfort Zone, by F. C. Houghten and C. P. Yaglou (A. S. H. V. E. Transactions, Vol. 29, 1923, p. 361).

nently fixed between the limits of 63 and 71 deg ET (normal), and the comfort line at 66 deg ET (normal). This explains the apparent discrepancy.

Another confusing matter which requires clarification is the range of comfortable humidity. In some comfort charts the average winter comfort zone has been extended along the 63 and 71 deg ET lines beyond the experimental limits to include the entire range of relative humidity from 0 to 100 per cent. The implication here is, of course, *not* that either the very dry or the very moist air conditions are entirely comfortable, but that they are equally warm.

In the comfort zone experiments of the A. S. H. V. E. Research Laboratory the relative humidity was varied between the limits of 30 and 70 per cent approximately, but the most comfortable range has not been determined. In similar experiments at the Harvard School of Public Health, a relative humidity of 70 per cent was found to be *somewhat humid* in winter, by about half of the subjects who were stripped to the waist, even when the dry-bulb temperature was 70 F or less. In summer, a relative humidity of 30 per cent was pronounced as *a little too dry* by about a third of the subjects wearing warm-weather clothing. So long as the temperature was kept within proper limits, the majority of the subjects were unable to detect sensations of humidity (*i.e.,* too high, too low, or medium) when the relative humidity was between 30 and 60 per cent. This is in accord with studies by Howell,[8] Miura,[9] and others.

Until more exact information is secured, it would be desirable to restrict the comfort zones to the range of relative humidity employed in the comfort zone experiments, namely 30 to 70 per cent. Relative humidities below 30 or slightly over 70 may prove satisfactory from the standpoint of comfort, so long as extreme humidities are avoided. From the standpoint of health, however, the consensus seems to favor a relative humidity between 40 and 60 per cent. In mild weather such comparatively high relative humidities are entirely feasible, but in cold or sub-freezing weather they are objectionable on account of condensation and frosting on the windows. They may even cause serious damage to certain building materials of the exposed walls by condensation and freezing of the moisture accumulating inside these materials. Unless special precautions are taken properly to insulate the affected surfaces, it will be necessary to reduce the degree of artificial humidification in sub-freezing weather to less than 40 per cent, according to the outdoor temperature. The principles underlying humidity requirements and limitations are discussed more fully elsewhere.[2-10]

The comfort chart (Fig. 2) embodies all the foregoing recommendations of the committee and is submitted to the Society for use as a tentative standard until more exact information is secured. The variation in the sensation of comfort within the zones is indicated by the comfort scales which give the percentage of subjects feeling comfortable at the various air conditions.

The extreme winter comfort zone extends from a minimum effective temperature of 60 deg at which all the subjects of the experiments were too cold,

---

[8] Humidity and Comfort, by W. H. Howell (*The Science Press*, April, 1931).
[9] The Effect of Variation in Relative Humidity upon Skin Temperature and Sense of Comfort, by U. Miura (*American Journal Hygiene*, Vol. 13, 1931, p. 432).
[10] Humidification for Residences, by A. P. Kratz (University of Illinois *Engineering Experiment Station Bulletin* No. 230, July 28, 1931).

to a maximum of 74 deg, at which all the subjects were too warm. The average winter comfort zone shown by the area shaded with thin solid lines, is included between the 63 and 71 deg ET lines. In a similar way it can be seen that the limits of the extreme summer comfort zones are 64 and 79 deg ET, and those of the average comfort zone, shaded by thin broken lines, 66 and 75 deg ET. The variation from winter to summer is probably due to adaptation to seasonal weather as well as to seasonal variation in the clothing worn.[11]

The comfort lines separate the cool air conditions on the left from the warm air conditions on the right. Under the air conditions existing along or defined by the comfort lines, the body is able to maintain thermal equilibrium with its environment, with the least conscious sensation to the individual, or with the minimum physiologic demand on the heat regulating mechanism. This environment involves not only the condition of the air with respect to temperature and humidity but also the condition of the surrounding objects and wall surfaces.

The comfort chart may be reproduced with the zones superimposed on the standard psychrometric chart or on any other suitable chart, but the essential features with respect to air movement and relative humidity limitations should remain unaltered.

Application of Comfort Chart

The average winter comfort line (66 deg ET) applies to average American men and women living inside the broad geographic belt across the United States, in which central heating of the convection type is generally used during four to eight months of the year. It does not apply to rooms heated by radiant energy. In so far as the committee is aware, the Society has never advocated the use of the chart in foreign countries where the climate, heating methods, and general living conditions are materially different from those of this country, though several foreign workers have attempted to show that it cannot be so applied. Even in the warm south and southwestern climates, and in the very cold north-central climate of the United States, the comfort chart would probably have to be modified according to climate, living and working conditions, and the degree of acquired adaptation.

In densely occupied spaces, such as class rooms, theaters, auditoriums, and the like, somewhat lower temperatures are necessary than those indicated by the comfort line on account of counter radiation between the bodies of occupants in close proximity. In rooms in which the average wall surface temperature is considerably below the air temperature, higher air temperatures are necessary. The reverse holds true in radiant or panel heating methods.

The sensation of comfort, insofar as the physical environment is concerned, is not absolute but it varies considerably among certain individuals. Therefore in applying the air conditions indicated by the comfort line the ventilating engineer should not expect all the occupants of a room to feel perfectly comfortable. When the winter comfort line is applied in accordance with the foregoing recommendations, the majority of the occupants will be perfectly comfortable, but there will always be a few who would feel *a bit too cool* and

[11] The Summer Comfort Zone: Climate and Clothing, by C. P. Yaglou and Philip Drinker (A. S. H. V. E. Transactions, Vol. 35, 1929, p. 269).

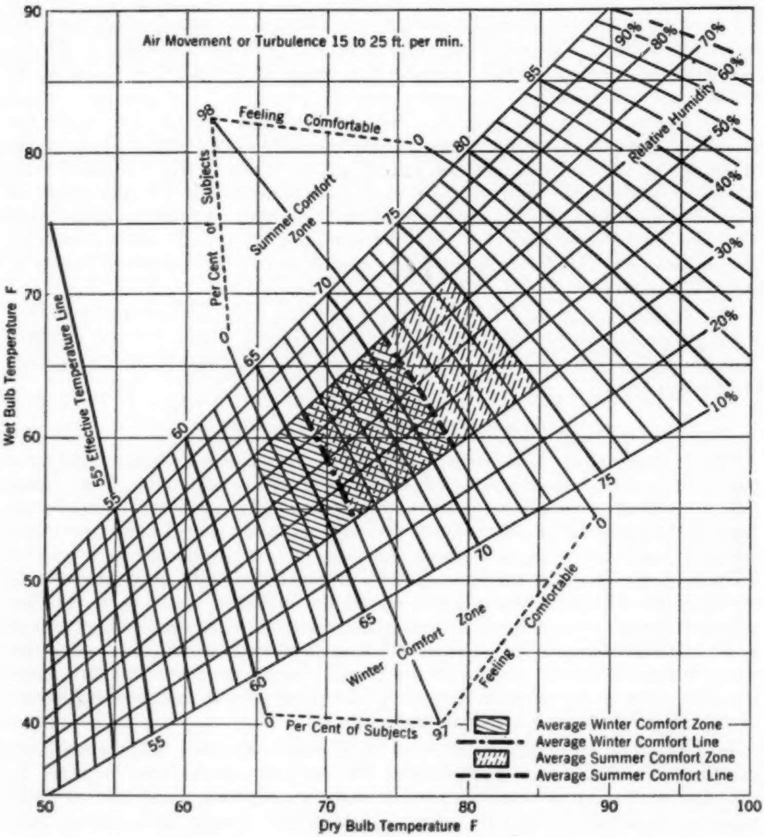


FIG. 2. COMFORT OR EFFECTIVE TEMPERATURE CHART FOR AIR VELOCITIES OF 15 TO 25 FPM (STILL AIR)

Note: Both summer and winter comfort zones apply to inhabitants of the United States only. Application of winter zone is further limited to rooms heated by central station systems of the convection type. The zone does not apply to rooms heated by radiant methods. Application of summer comfort zone is limited to homes, offices and the like, where the occupants become fully adapted to the artificial air conditions. The zone does not apply to theaters, department stores, and the like where the exposure is less than 3 hours.

a few *a bit too warm*. These individual differences among the minority may be counteracted by suitable clothing.

It is sometimes argued in the literature that air conditions lying outside the comfort zone have been found perfectly comfortable to certain persons, in cases in which the use of the comfort chart is advocated. If these claims are examined, it will be found that in the majority of them, if not in all, the authors did not appreciate the fact that they were referring to the *average* comfort zone, within which 50 per cent or more of the occupants of a room are expected to be comfortable. In other words, it is possible for half of the occupants of a room to be comfortable in air conditions outside the *average* comfort zone, but in the majority of cases, if not in all, these conditions will be well within the extreme comfort zone as determined experimentally.

Strictly speaking, the only authoritative comfort zone on which accurate data are available is that for 15 to 25 fpm air movement or turbulence (often referred to as still air). In the past, the winter comfort zone has often been superimposed on the thermometric chart or on effective temperature charts for various air velocities, on the assumption that air conditions of equal warmth are approximately equally comfortable. This may hold in hot industries where the workers are adapted to high temperatures and strong air currents, but it does not apply to sedentary conditions such as are found in homes, offices, theaters, and the like, and therefore, the unqualified use of such charts is not advocated. To avoid misunderstanding, these charts should be redrawn with the comfort zones omitted. If one is interested in ascertaining approximately whether a given industrial condition is reasonably comfortable, it would be necessary first to compute the effective temperature from the thermometric chart (Fig. 1) and then refer this effective temperature to the comfort or effective temperature chart (Fig. 2).

The summer comfort line (71 deg ET) is applicable to the same geographic area as the winter comfort line. It is further restricted to cases in which the human body has reached thermal equilibrium with its environment. As a general rule this takes place after 1½ to 3 hr exposure. When a person from outdoors enters a room cooled to 71 deg ET on a hot day (95 F or over) an intense chill is likely to be experienced which is unpleasant. However, after remaining in the room for about 2 hr, this fundamental optimum condition will prove satisfactory to the average person. The summer comfort zone, as well as the comfort line, makes proper allowance for these adaptive changes in the body, and thus applies to homes, offices, schools and other similar places where persons of sedentary occupations spend from 3 to 8 or more hours daily.

In artificially cooled theaters, department stores, restaurants, and other public buildings where the period of occupancy is short, the contrast between outdoor and indoor air conditions becomes the deciding factor in regard to the temperature and humidity to be maintained. The object of cooling such places in the summer is not to reduce the temperature to the optimum degree, but to maintain therein a temperature which is temporarily comfortable to the patrons who thus avoid sensations of chill and intense heat on entering and leaving the building. The relative humidity should be low enough (about 50 per cent) to give a sense of comfort without chill and to induce a rate of evaporation which will keep clothing and skin dry. For exposures less than 3 hr, desirable

TABLE 1. DESIRABLE INDOOR AIR CONDITIONS IN SUMMER CORRESPONDING TO OUTDOOR TEMPERATURES

Applicable to Exposures Less Than 3 Hours

Outdoor Temp. (Deg Fahr)	Indoor Air Conditions with Dew-Point Constant at 57 F		
Dry-Bulb	Dry-Bulb	Wet-Bulb	Effective Temp.
95	80.0	65.0	73
90	78.0	64.5	72
85	76.5	64.0	71
80	75.0	63.5	70
75	73.5	63.0	69
70	72.0	62.5	68

indoor conditions in summer corresponding to various outdoor temperatures are given in Table 1.

The comfort zone and comfort line for men normally clothed and working at the rate of about 33,000 ft-lb per hour are discussed elsewhere.[12]

DISCUSSION

HAROLD L. ALT (WRITTEN): It is gratifying to note year by year the progress being made along the most important line of endeavor towards human comfort. The very elusiveness of the matter of comfort makes it doubly intriguing to search for that narrow causeway of temperature and humidity combination which combined with the other factors existing in a building in summer or winter will give the occupants the maximum comfort sensation. It has been recognized for some time that the summer comfort zone does not coincide with the winter comfort zone due to the fact that the human body is a wonderful machine and surprisingly accommodates itself to changed outside conditions such as winter's cold or summer's heat.

Comfort constantly reminds one of the immortal Lincoln's saying that you can't fool all of the people all of the time and in comfort zone work it seems apparent that you can't make all of the people comfortable all of the time even when conditions are held exactly on the comfort line to say nothing of being simply in the comfort zone. Comfort's highest batting average cannot exceed 97 per cent and is much more likely to be around 60 to 70 per cent so that the complaint of one or two persons as to conditions means exactly nothing, provided that the others are satisfied.

It does seem however that there should be more uniformity in determining what the best inside conditions of a building should follow than there is; for example, some time ago before this summer comfort zone was available, the case of cooling a large building was under consideration and the writer has tentatively decided on 80 dry bulb with about 50 per cent relative humidity during the extreme weather which gives an effective temperature of 74 F and which looks fairly good on the summer comfort zone indicated in this paper but two other companies who are making a specialty in cooling installations came in with recommendations that were quite different, one suggesting and recommending 85 F dry bulb and 40 per cent relative (with a resultant effective temperature of 76½ F) while the other was in favor of 60 per cent relative but a dry bulb of 75 F (giving an effective temperature of 71 F).

[12] Heat and Moisture Losses from Men at Work and Application to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant (A. S. H. V. E. TRANSACTIONS, Vol. 37, 1931, p. 541).

If these three sets of conditions are plotted on the summer comfort zone chart it will be found that the conditions assumed by the writer are almost exactly half way between the recommendations of the other two organizations both in regard to dry bulb, relative humidity and effective temperature. The summer comfort zone does show on the chart, however, that the 75 F dry bulb and 60 per cent relative humidity fall exactly on the comfort line for summer indicating that this recommendation really was the best of the three. Yet, if this is so, there will be altogether too much of a temperature shock for those entering from the outside air and the writer believes that no building should be kept over 10 deg effective temperature below the outside to avoid the danger of bad effects on those entering in a hot and perspiring condition from the street.

Malcolm C. W. Tomlinson (Written): The preparation of research data for standardization should cover two points frequently omitted. Can the terms used be simplified and how can the information be put into the language of the non-technical individual? In a world struggling to free itself from a serious depression the practical application of the infant art of comfort depends largely on how clearly we tell the story of what it is and what it can do. No matter how good it is we certainly will never sell it to the general public unless we explain it in terms that can be understood.

Now the main difficulty is that the terminology of all research products is created by scientists who speak a complex language. No more vivid example of the disastrous effect of the terms they use, outside of air conditioning, need be sought. Just recollect the difficulties of explaining humidity, humidification and dehumidification. If these few terms had been simplified, if they had been expressed in the common language, would not air conditioning have made greater strides?

The creation of a committee looking towards standardization of comfort data affords an opportunity for the consideration of comfort terminology standardization. The difficulty is that, to obtain best results, such a committee should contain members who are not scientifically trained but who are versed in the construction of words.

The need of proper terminology in comfort will be found, for example, in the expression *effective temperature*. For an index which covers the combined effects of relative humidity, temperature and air motion is it reasonable to expect the public to understand a term which uses the word temperature? How can we help them to separate it, in their minds, from the ordinary word temperature which we so carefully designate as dry bulb temperature? It may be argued that this index refers solely to heating and cooling effects. The reply to that viewpoint could easily be that a heat unit may be just as satisfactory a term as temperature. Why, then, not set up some special form of heat unit to express warmth and cold?

Again consider the fact that the so-called effective temperature lines are not, actually, lines of equal comfort. Yet we still term our chart a comfort chart and, as long as we use the word comfort for the condition produced by scientifically controlled weather on the human body, most people using our data will continue to consider such lines as lines of equal comfort.

These few examples should point the way to the need for a careful examination of all comfort terminology.

It is gratifying to note the limitation of health placed on the comfort data. Probably a better term would be normal health. It is doubtful if many human beings enjoy perfect health but no doubt many are in a condition of normal health so that their reactions to variations in weather are average.

In connection with the proposed zone limitations of from 30 to 70 per cent relative humidity on the comfort chart it is possible, when experimental evidence is had in

the low humidity field, that the direction of the effective temperature lines in such regions will have to be changed. The writer has found that higher temperatures than those indicated by the comfort charts must be used to obtain comfort in dry atmospheres between 10 per cent and ¼ per cent relative humidity.

The use of the thermometric chart is no doubt advocated because it eliminates the need for a number of comfort charts at various air movements. Nomographic or alignment charts of this nature are not very handy so it is to be hoped that the committee will find a more satisfactory method of determining the effective temperature.

THOMAS CHESTER: This paper is of value to the Society because it reveals present knowledge relating to comfort as affected by atmospheric conditions, in a manner which can be more easily understood than the previous disclosures.

It should be regarded merely as an interim report, as a great deal remains to be learned on this subject. The information given about the winter comfort zone can be regarded as quite satisfactory for the present. The same thing cannot be said about the information given concerning the summer comfort zone. It is generally known that a large number of air conditioning installations designed for producing summer comfort, were based on wrong conceptions. Practically all the earlier cooling systems overcooled the indoor atmosphere and maintained excessive relative humidities.

It seems evident that the misconceptions about summer comfort have not entirely disappeared. Fig. 2 shows the summer comfort zone as extending between the 70 per cent and 30 per cent relative humidity lines. It would be better to make the relative humidity range from 60 per cent to 30 per cent, as 70 per cent seems too high.

The optimum summer effective temperature is given as 71 deg, but I am of the opinion that this should be changed to 72.5 deg.

There is unnecessary vagueness about the application of the comfort chart, together with some qualifying remarks about possible modifications in requirements in accordance with local climatic conditions. It would seem better to definitely state that the chart covers comfort conditions in the United States for 42 deg North Latitude. For each reduction of 1 deg in North Latitude, the optimum Effective Temperature should be shifted 0.25 deg to the right on the Comfort Chart.

On this basis New Orleans, which is at 30 deg North Latitude, should have a summer optimum effective temperature of 74 deg if the present 71 deg optimum is adhered to for 42 deg North Latitude. If the Chart optimum is raised to 72.5 deg Effective Temperature for 42 deg North Latitude as recommended, then the summer optimum Effective Temperature for New Orleans should be 75.5 deg.

Summer heat depends essentially upon the comparative elevation of the Sun, which is vertically over head at the Tropic of Cancer on June 21. Maximum summer heat occurs about one month later than maximum solar elevation due to accumulation of

Degrees Fahrenheit, Outside	Degrees Fahrenheit, Inside			
Dry-bulb	Dry-bulb	Wet-bulb	Dew-point	Effective Temperature
100	82.5	69.0	62.3	76.0
95	81.0	67.7	60.8	74.8
90	79.5	66.5	59.5	73.6
85	78.1	65.3	58.0	72.5
80	76.7	64.0	56.6	71.3
75	75.3	63.0	55.6	70.2
70	74.0	62.0	54.5	69.0

heat in the atmosphere. It is therefore logical to determine definite Effective Temperatures in summer in accordance with Latitude.

In addition to the summer optimum Effective Temperature being too low as already stated, the temperatures given in Table No. 1 relating to "Desirable Indoor Conditions in Summer corresponding to Outdoor Temperatures" are too low and I think they should be amended as noted in the accompanying tabulation.

H. W. Schmidt: We are specifically interested in using the comfort chart for schoolhouse conditions and air conditioning of such structures. A question has come up during the last two or three years which may be of interest, and that is the question as to how far the subjective reactions of the adult, which are of course fundamental and basic in this study, differ from the same reactions of the child. I don't know whether we are specifically warranted in accepting the empirical findings of the chart and utilizing them directly with the child for a number of reasons which probably have already occurred to you.

Of course, there are sufficient variations and limitations in the comfort zone to probably permit the adaptation of the child organism, which is extremely adaptable to varying conditions as they occur in schoolroom procedure, but we do know that the respiration rate of the child and the temperature metabolism differ from that of an adult. We know that a child's physical reactions are somewhat different from the adult and the question that has come before us on a number of occasions in the last few years is: whether it is advisable, feasible or necessary to find out just what the child's reaction is. Children ranging in age from 10 to 15, would probably give specific and warrantable reactions when subjected to conditions of the comfort zone and temperature charts that we are using without question.

We are looking for light on this subject and I do not know of any experiments, except those of the New York Commission, which have attempted to use the child's reaction, I think and believe, in a rather unscientific way. There is the possibility that the comfort chart may be changed or may include a lower or upper range when applied to schoolroom conditions on the basis of child reaction as compared to the same effect when we are using them with adults, and on the basis of which the chart has been developed. This information is important, especially today under the conditions under which we have to heat and ventilate schoolhouses where I am afraid the economic pressure is beginning to supersede all other considerations.

Mr. Carrier: I think that the summary that Mr. Houghten has presented is of vital interest to every one in the heating and air-conditioning work. The data taken are for adults but are very enlightening when considered in connection with the work done subsequently on the rate at which heat is given off from the human body.

The sense impression with reference to comfort is probably shown to have no direct relationship to the body metabolism by the subsequent research work done after this chart was completed. We had thought in proposing these later tests that we would get another measure, a physiological measure, if you please, of the comfort lines. The surprising thing was that practically any point within the zone of comfort gave almost precisely the same total body metabolism. In other words, under normal conditions the body metabolism was not affected greatly. It was entirely a question of skin sensation, perhaps a sensation due to the warmth or temperature of the skin surface which was rather difficult to measure because it varied so considerably over different parts of the body.

While the Laboratory and the sponsors for the work in the Committee on Research failed in accomplishing their object of establishing another basis for effective temperature determination, they did add some very interesting information with reference to the body reactions under different temperature and humidity conditions.

This is a line of research that could not be well undertaken by any private organi-

zation. It is not for the purpose of developing any type of equipment, either heating equipment or air-conditioning equipment, but it is universally available for the entire industry represented by our organization, whether engineers, or manufacturers of equipment, in determining the standards to which their equipment should be designed or manufactured. It is a line of investigation that is universally valuable to the whole art. I believe that our work in research should follow these broad lines of basic requirements in the industry in preference to any narrower investigations of particular or special equipment or equipment applications. It is to be realized that some of this work is necessary, but that work can well be done by private laboratories. There are developments in every company towards special products. The function of this Society is perhaps as well illustrated by this line of research which I believe has already become extremely valuable to the whole industry and in the future will be appreciated to a much greater extent than at present.

MR. NESBITT: Like Mr. Schmidt, I would be very much interested in knowing if data are available showing the amount of heat given off by children of different ages as compared to the amount of heat given off by an adult. I understand, of course, that the amount of heat given off is a function of the skin surface, but is it the same for a 10-year old child as it would be for a 40-year old man per unit of surface, if they were both doing the same kind of work?

MR. EWALD: The Chairman mentioned one item in the tests, that an Englishman had preferred a lower temperature than some of the people of other races, which brings a thought that possibly the races which originate in the southern countries would desire a different comfort zone than those of the northern races. Perhaps the Spaniards and Italians would not be comfortable in the same temperature that the Scandinavians would. To carry the thought a little further, perhaps the negro race would be comfortable at a slightly higher temperature than the white race. I don't know that any such investigation has been included in the report.

MR. HOUGHTEN: The Laboratory does have available additional data not entirely completed on the child, which will be published. These data, while not as complete as that for adults, show that the school child from his own desire chooses a slightly higher temperature than the adult, not enough to make very much difference. It is, however, on the higher side rather than the lower side.

The Laboratory also has some data available that will be published in the near future on metabolism and heat loss, heat dissipation in the atmosphere, for the child of school age and various ages, which again show that the metabolic rate of the child while considerably different from the adult as a total when taken in terms of metabolic rate per unit surface area of the body is not very different. For children of six and seven years and younger there is a slight increase in the metabolic rate per unit surface area, but for practical purposes and for the purpose of the air-conditioning engineer all these processes metabolic rate, heat dissipation, heat production in the body, are proportional to the surface area of the individual.

There is a very important question involved in what we speak of as the *ideal temperature*. That ideal temperature was determined by vote of subjects in the tests on adults and again it was so done for children of school age.

Ten years ago there was a very different basic opinion held by Dr. E. V. Hill and myself. At that time I would have said that we were better off if we chose temperatures slightly below that at which we were comfortable. In other words, if we went down the scale a little bit we were better off. Dr. Hill most decidedly believed and emphasized that probably the reverse was true, that it was rather a psychrometric point of view, that those things which were not entirely comfortable to us were better for us. Recently we had the opportunity at the Laboratory to get a little data on that point. The Pennsylvania State Board of Education is making a study of temperatures desired in the school, including swimming pools, and they had some

studies made in our Laboratory one day to see what temperature the atmosphere should be in a swimming pool room to be best for the child. A few children were put in a psychrometric room at the Laboratory with bathing suits on and they were wetted down by sprays of water at a temperature that the swimming pool would be and allowed to do what they wanted in those rooms. We got some very interesting data. When the child was cold, decidedly cold, he huddled together and did not want to be active. When he was a little bit on the warm side of the comfort line he was very active; he was out and wanted to play all the time. When we got a little higher, so the temperature was decidedly too warm, again he wanted to become languid and lie down and be inactive, which in my mind at least bears out the earlier opinion of Dr. Hill that we are all better off in all of our activity on the slightly warm side of the comfort line than on the cold side. I am now of that same opinion.

Another question was brought up regarding the application of the comfort zone to people of different racial characteristics and from different geographical regions, particularly latitudes. That is an important question and one which we do not have much data on. Prof. Yaglou has given some study to that through surveys and inquiry and the best information available is that these data apply particularly to people in this part of the country, neither as far north as the most northern parts of the United States, nor the southern parts. Also it applies particularly (and this we do have data on) to activity and characteristics as regards clothing of the average American in this region which may be different from other parts of the country.

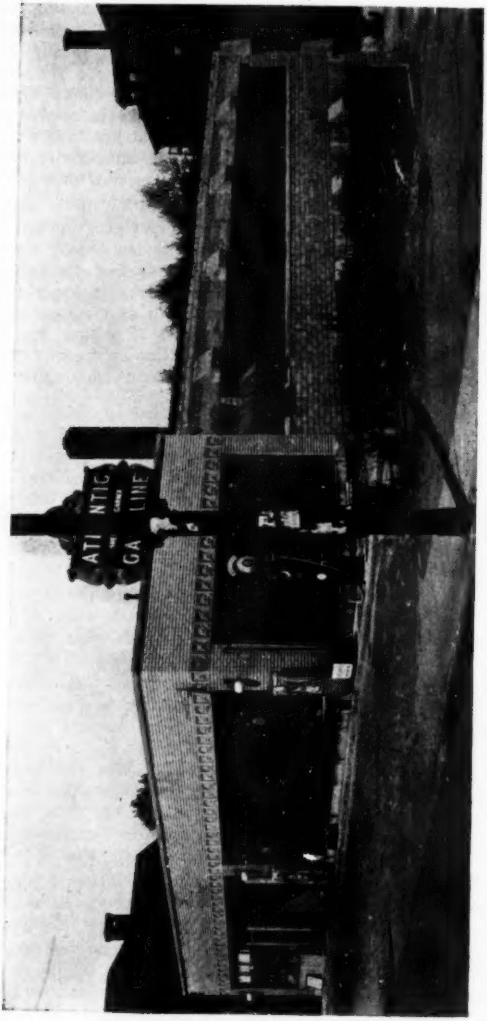


FIG. 1. SHERADEN GARAGE, SHERADEN, PENNSYLVANIA

No. 934

CARBON MONOXIDE DISTRIBUTION IN RELATION TO THE VENTILATION OF A ONE-FLOOR GARAGE

By F. C. Houghten [1] (*MEMBER*) and Paul McDermott [2] (*NON-MEMBER*), Pittsburgh, Pa.

THE development of the Code for Heating and Ventilating Garages,[3] adopted by the Society in 1929, made apparent a need for additional facts and information concerning the life and fire hazards in garages resulting from carbon monoxide and other physiologically harmful or combustible gases or vapors found in garages, and means for satisfactorily and economically eliminating such hazards. A Technical Advisory Committee of the Committee on Research of the Society was organized in 1928 to study the needs for research in this field.

The Technical Advisory Committee on Ventilation of Garages and Bus Terminals, under the chairmanship of E. K. Campbell, has been actively engaged in planning this work. A brief study was made at Washington University, St. Louis, Mo., in cooperation with the A. S. H. V. E. Research Laboratory, the results of which were reported in a paper entitled, Carbon Monoxide Concentration in Garages,[4] by A. S. Langsdorf and R. R. Tucker. This study was made in a large commercial garage equipped with means for mechanical ventilation, but usually operated with natural ventilation only. The lack of success experienced by the investigators in controlling ventilating conditions, including the opening and closing of doors and windows, emphasized the need for additional studies in garages where the investigators are free to control the means, method and volume of ventilation, and where they are otherwise free to control the entire surroundings in accordance with the requirements of the study.

In the fall of 1931, the A. S. H. V. E. Research Laboratory was authorized

---

[1] Director, Research Laboratory, American Society of Heating and Ventilating Engineers, Pittsburgh, Pa.

[2] Research Engineer, Research Laboratory, American Society of Heating and Ventilating Engineers, Pittsburgh, Pa.

[3] See A. S. H. V. E. Transactions, Vol. 35, 1929, p. 355.

[4] See A. S. H. V. E. Transactions, Vol. 36, 1930, p. 511.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Milwaukee, Wis., June, 1932, by F. C. Houghten.

to proceed with a study in Pittsburgh. For this purpose, the small single room, window ventilated, Sheraden Garage located in Sheraden, a suburb of Pittsburgh, and also the five-story, underground ramp garage in the basement of the Grant Building were made available under conditions highly satisfactory and helpful to the investigation. Studies were made in these garages early in 1932. The results of the study in the Sheraden Garage are the basis of this paper, while those from the Grant Building Garage are published in a separate paper.

SHERADEN GARAGE

The Sheraden Garage (Figs. 1 and 2) is a two-story building with the upper floor at the street level in front, and the lower or basement floor slightly above ground in the rear. The upper floor is a solid concrete slab with no openings between the basement and the upper level, except through the door, *A*, leading to a stairway to the lower floor. Hence, the second floor of the garage is entirely independent of the basement floor as far as ventilation is concerned. The study was made in the upper floor of this building.

The upper floor of the garage consists of one large room or garage space, 60½ ft wide and 52 ft long. The distance from the floor to the concrete ceiling slab is 11 ft 8 in. Reinforced concrete beams, *B* and *C*, extend down from the ceiling slab 27 in. and 20 in., respectively. The entrance to the floor is by an overhead door, *D*, 8 ft wide and 9 ft high, leading into the garage space by an alley or hallway, on one side of which is an automobile showroom, and on the other side, a small storeroom. Both the first and second floors of the garage are heated by unit heaters, which were not in operation during the test.

There is no provision for mechanical ventilation, but three sides of the room are lined with horizontally-pivoted windows which open as shown by *E*, Fig. 2. Mechanical ventilation was provided for the study by arranging six 15-in. propeller fans into respective boxes which were fitted into openings made by removing the upper panes of glass from six windows. Three of these boxes, *F*, *G*, and *H*, were distributed along the right hand wall, and three, *I*, *J*, and *K*, along the left hand wall. These boxes were so arranged that the fans could be used for either supplying or exhausting air. The boxes were so designed and installed that the air was supplied or exhausted through the top of the box. Hence, the air was supplied to or taken from the garage at a point within 1 ft of the ceiling, as shown in the elevation view by fans *I* and *J*.

For the purpose of admitting air when the fans were operating as exhaust, or as a vent when the fans were operating as supply, the box, *L*, was built over window, *E*. The only opening from the room into this box was at the bottom about 1 ft from the floor, so that air was either admitted to or exhausted from the garage at the floor line as indicated. Air was also admitted to or vented from the building at the front by partially opening the overhead door, *D*, and sealing off the upper crack resulting from the partially opened door.

This arrangement admitted air at the front and rear at the floor line, and allowed it to escape by gravity through open windows or by forced exhaust through the fans. It also allowed for forced air supply at the ceiling by the fans, in which case the air left the garage at the front and rear floor line. Cross ventilation could also be provided by making the fans on one side

supply air to and those on the other exhaust air from, the building. When exhausting air from the building, each fan installed in its box handled 1,100 cfm of air. When supplying air to the building, each fan installed in the boxes delivered 800 cfm of air.

The temperature in the garage was approximately 60 F at the time of beginning the surveys on both nights. During the surveys, while the cars were idling, the temperature rose at times as much as 5 deg. The outdoor temperature on both nights was approximately 55 F. The first night during which the study was made (April 16, 1932), there was a wind movement of about 5 mph from the right; *i.e.*, blowing against the side of the garage next to the fans, *F*, *G*, and *H*.

PURPOSE OF STUDY

The purpose of the study was essentially to determine facts of value in the ventilation of garages. Since concentration of carbon monoxide resulting from the combustion of gasoline in cars is the controlling factor affecting the ventilation of garages, the investigation became a study of carbon monoxide concentrations. In this connection, it was desirable to determine the quantity of air necessary to keep the concentration of carbon monoxide within safe limits, depending upon the garage characteristics and car activity. These facts should be of value in estimating the required quantity of air to be handled in any garage ventilation problem, and in arriving at the best point for supplying and exhausting air.

METHOD OF ATTACK

The study resolved itself into a series of surveys designed to give the concentration of carbon monoxide within the garage, depending upon the type and amount of ventilation and the car operation. The most convenient method of making such surveys, provided the concentration of carbon monoxide is greater than one part per 10,000, and provided the variation in concentration sought for is not less than one part in 10,000, is the *pyrotannic acid method*, developed by the U. S. Bureau of Mines.

This method of determining concentrations of carbon monoxide requires the collection of 250 cc samples of air. The standard practice is to use an ordinary rubber bulb aspirator and pump the sample into a 250 cc rubber stoppered bottle. Fifty squeezes of a 75 cc rubber bulb have been proved sufficient to give a representative sample. The samples contained in the 250 cc bottles are then taken to a laboratory for the analysis, which is completely described in a U. S. Bureau of Mines publication.[5] Briefly, the analysis consists of absorbing the carbon monoxide contained in a sample into 0.1 cc of human or ox blood diluted to 2 cc with distilled water, and comparing the resulting color, after chemical treatment with pyrotannic acid, with a standard set of colors. With proper equipment, a trained observer can analyze 25 air samples per hour. The pyrotannic acid method of analysis has a distinct advantage over other methods in that a large number of samples of air can be taken

[5] The Pyrotannic Acid Method for the Quantitative Determination of Carbon Monoxide in Blood and Air, by R. R. Sayers, W. P. Yant and G. W. Jones. Report of Investigations, Serial No. 2486, Bureau of Mines, June, 1923, 6 pp.; Technical Paper 373, Bureau of Mines, 1925, 18 pp.

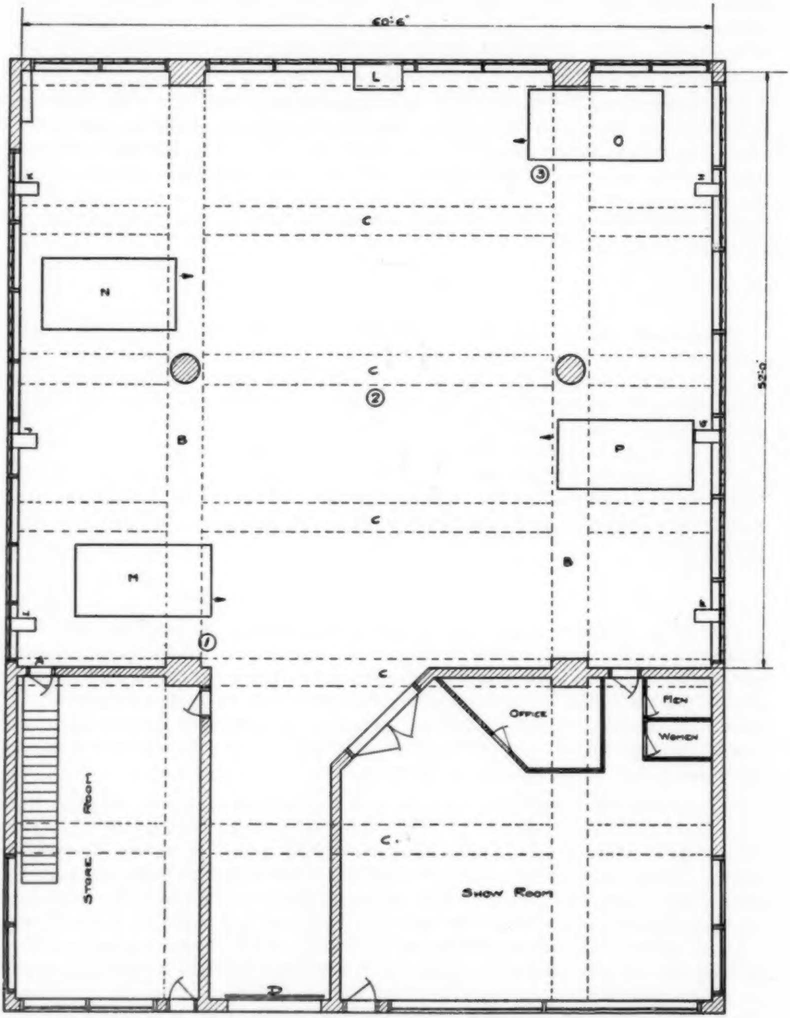


FIG. 2. SHERADEN GARAGE, SECTIONAL DRAWING

A—Door to Basement
B—Reinforced Concrete Beams, 27 in. x 36 in.
C—Reinforced Concrete Beams, 20 in. x 30 in.
D—8 ft x 9 ft Door Entering Garage
E—Pivoted Window
F, G, H, I, J, K—Fan Boxes
L—Box Over Window *E*
M, N, O, P—Positions of Idling Cars
①, ②, ③—Sampling Stations

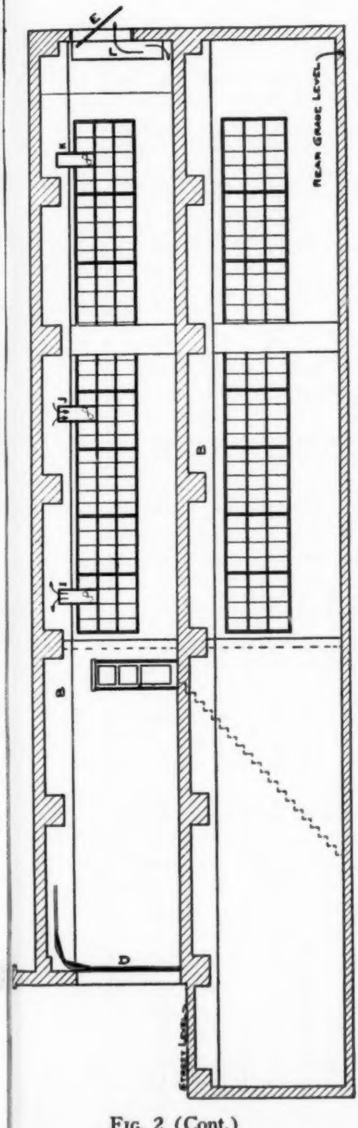


Fig. 2 (Cont.)
(See opposite page for key)

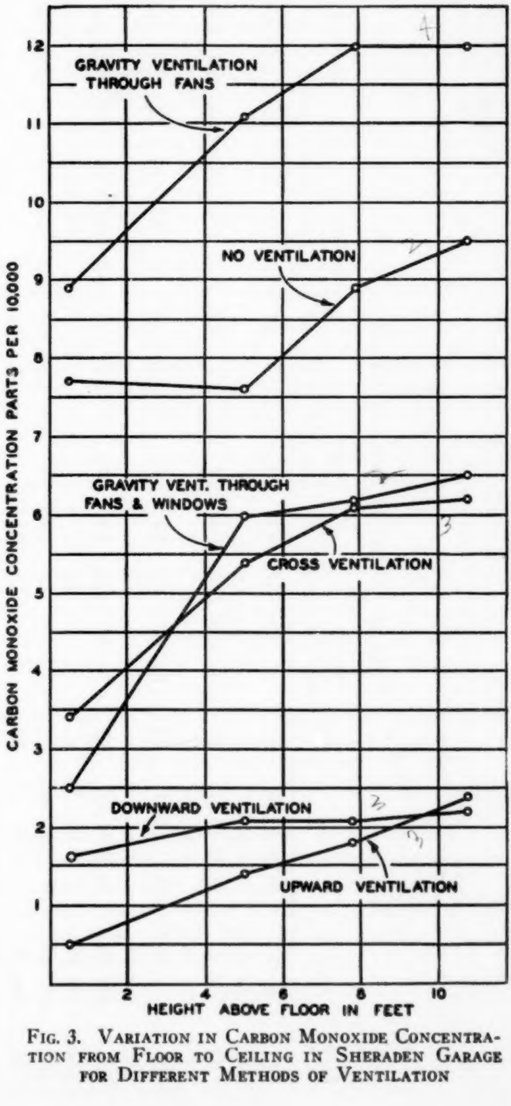


Fig. 3. Variation in Carbon Monoxide Concentration from Floor to Ceiling in Sheraden Garage for Different Methods of Ventilation

TABLE 1. SURVEYS IN SHERADEN GARAGE

Survey No.	Date	Time	Station 1				Station 2				Station 3				Av. Sta. 1, 2, 3			
			6"	5'-0"	7'-8"	10'-8"	6"	5'-0"	7'-8"	10'-8"	6"	5'-0"	7'-8"	10'-8"	6"	5'-0"	7'-8"	10'-8"
29	Apr. 16	10:00–10:06	8.0		8.0	6.7	9.4	9.4	12.0	15.0	5.7	5.7	6.7	6.7	7.7	7.6	8.9	9.5
29	Apr. 16	P.M.				7.0												
30	Apr. 16	10:28–10:31	8.0	12.0	12.0	12.0	9.4	12.0	12.0		9.4	9.4	12.0	12.0	8.9	11.1	12.0	12.0
30	Apr. 16	P.M.				12.0												
31	Apr. 16	10:41–10:44	0.3	5.7	6.7	6.7	3.8	6.7		6.7	3.3	5.7	5.7	5.7	2.5	6.0	6.2	6.4
31	Apr. 16	P.M.				6.7												
32	Apr. 16	10:57–11:02	4.7	6.7	8.0	6.7	3.8	5.7	5.7	5.7	3.3	3.8	4.7	4.7	3.9	5.4	6.1	6.2
32	Apr. 16	P.M.				9.4												
33	Apr. 22	11:35–11:40	1.9	1.9	2.5	2.5	1.9	2.1	2.5	2.5	0.8		1.3	1.9	1.7	2.0	2.1	2.4
33	Apr. 22	P.M.	2.5			2.5	2.1			3.0	0.8							
34	Apr. 22	11:48–11:52	1.3	1.9	2.5	2.5	1.9	2.5	2.5	1.3	0.8	2.5	1.3	1.9	1.6	2.2	2.1	1.9
34	Apr. 22	P.M.	1.9			2.5	1.9			1.3	1.9							
Average Survey 33, 34			1.9	1.9	2.5	2.5	1.9	2.3	2.5	2.0	1.1	2.5	1.3	1.9	1.6	2.1	2.1	2.2
35	Apr. 23	12:56–1:00	0.3	0.8	1.3	1.9	0.8	0.8	2.5	1.9		1.9	2.1	2.5	0.6	1.2	2.0	2.3
35	Apr. 23	A.M.	0.8			2.5	0.3			2.5				2.5				
36	Apr. 23	1:10–1:14	0.3	1.9	1.9	2.5	0.3	0.8	0.8	2.5		1.9	1.9	2.5	0.4	1.5	1.5	2.5
36	Apr. 23	A.M.	0.0			2.5	0.8			2.5				2.5				
Average Survey 35, 36			0.4	1.4	1.6	2.4	0.6	0.8	1.7	2.4		1.9	2.0	2.5	0.5	1.4	1.8	2.4
Average															3.4	4.6	5.1	5.4

over a short period of time, representing conditions at many points through the space surveyed.

Carbon monoxide as a hazard in respired air has been studied extensively by the U. S. Bureau of Mines, the U. S. Public Health Service and many other organizations.[6] While the investigators have been reluctant to recommend any concentration as safe, it is generally recognized that one part of carbon monoxide in 10,000 parts of air can be breathed indefinitely without noticeable injury. Slightly higher concentrations, possibly as high as two parts in 10,000, may be endured without noticeable injury for shorter periods of time. In many of these studies carbon monoxide concentrations have been recorded as parts in 10,000, which terminology has been fairly well standardized and will be used in this paper. Hence, when any numerical concentrations are spoken of, they will be understood to mean parts in 10,000.

SURVEYS IN THE GARAGE

Eight surveys were made in the Sheraden Garage. In all surveys, samples of air were taken for analysis by the pyrotannic acid method at points, 1 ft, 5 ft, 7 ft 8 in., and 10 ft 8 in., above the floor at Stations *1, 2,* and *3,* Fig. 2. In all of the surveys conditions of car operation, opening and closing of windows and doors, and ventilation were under complete control of the investigators. Cars in some of the positions, *M, N, O,* and *P,* Fig. 2, were allowed to idle. All spaces along the right and left hand walls were lined with cars parked similar to idling cars; there were, however, no cars in the middle space between beams, *B*. The conditions and results of the surveys, together with any conclusions to be drawn from the findings are listed below.

Survey 29, No Ventilation, April 16, 1932, 10:00 to 10:04 P.M.

All windows and doors were closed and no mechanical ventilation was provided for this survey. Four cars, *M, N, O,* and *P,* Fig. 2, were started idling at 9:50 P.M., and samples were collected between 10:00 and 10:04. The analyses of the samples are tabulated in Table 1. The average concentration of carbon monoxide found at the 4 elevations is plotted in Fig. 3, giving the curve labeled *no ventilation.*

An unsatisfactory condition existed at the time of this survey one sample indicating 15 parts of carbon monoxide per 10,000 parts of air, which concentration is extremely dangerous, except for very short exposures. In fact, the condition was such as to make physiological collapse possible in a 30-min exposure.

The visible smoke in the exhaust from the cars rose very noticeably and tended to stratify at the ceiling. However, by the time the cars had been in operation 4 or 5 min the stratification of visible smoke had lowered, so as to fill the entire garage, with little noticeable demarcation. Conditions in the garage at the time of the survey were visibly bad, and the smoke and other products of combustion in the exhaust gases irritated the throat and eyes.

Survey 30, Gravity Ventilation Through Fans, April 16, 1932, 10:28 to 10:31 P.M.

For this survey the conditions in Survey 29 were altered by reducing the

[6] See Bibliography in the bulletin, Carbon Monoxide Poisoning in Industry, issued in 1930 by the Department of Labor, State of New York, pages 231 to 238.

TABLE 2. AIR REQUIRED TO DILUTE AUTOMOBILE EXHAUST GASES TO SAFE PROPORTIONS IN AIR BREATHED

(Car tested under summer conditions unless otherwise stated)

Number of Cars Tested	Type of Car [a]	Speed Miles per Hour	Average Carbon Monoxide in Exhaust Gas		Cubic Feet per Minute of Ventilating Air per Car Required to Dilute Carbon Monoxide to 1 Part per 10,000 (safe for indefinite period)
			Per Cent	Cubic Feet per Hour per Car (at 65 F, 29.92 in. Hg)	
		Engine Idling			
5	5P	Car standing	7.3	39.2	6533
7[b]	7P	Car standing	6.3	35.3	5883
5[b]	1½T	Car standing	7.1	31.0	5167
4	1½-3	Car standing	2.4[c]	13.2[c]	2200[c]
3	3½-4½	Car standing	8.8	66.4	11070
6	5T	Car standing	6.8	50.5	8417
		Engine Racing			
5	5P	Car standing	7.1	76.9	12820
7[b]	7P	Car standing	7.8	137.0	22830
5[b]	1½T	Car standing	7.7	67.9	11320
4	1½-3	Car standing	4.7	55.8	9300
3	3½-4½	Car standing	8.6	158.2	26370
6	5T	Car standing	7.5	105.2	17530
		Average Load and Level Grade			
11[b]	5P	3	6.7	36.3	6050
7[b]	7P	3	6.6	53.1	8850
5[b]	1½T	3	7.8	47.4	7900
22	1½-3	6	7.3	67.6	11270
16	3½-4½	6	7.8	92.6	15430
18	5T	6	7.5	110.3	18380
9	5P	10	7.8	44.1	7350
6	7P	10	8.3	75.5	12580
7	1½T	10	7.6	58.7	9783
20	1½-3	10	7.5	104.0	17330
13	3½-4½	10	7.5	147.6	24600
12	5T	10	6.9	151.6	25270
9	5P	15	6.8	57.9	9650
6	7P	15	8.6	111.8	18630
7	1½T	15	6.7	77.2	12870
15	1½-3	15	6.2	103.9	17320
5	3½-4½	15	4.8	131.2	21870

[a] Types are designated thus: *5P* and *7P* represent 5 and 7 passenger cars, *1½T*, 1½-3, 3½-4½ and *5T* represent trucks of the respective tonnages.
[b] Tested under winter conditions.
[c] These averages are undoubtedly too low to represent correctly the type of trucks. All cars were tested with carburetors set as submitted by the car owners. These four happened to be set for lean mixtures.

number of idling cars from 4 to 2 (cars *N* and *P* were shut off), uncovering the top opening of the fan boxes so that air could pass out through the fans by gravity, raising the front door 20 in. above the floor and opening the window, *E*, in box *L*. These changes were made at 10:05. The same number

TABLE 2 (Cont.). AIR REQUIRED TO DILUTE AUTOMOBILE EXHAUST GASES TO SAFE PROPORTIONS IN AIR BREATHED

(Car tested under summer conditions unless otherwise stated)

Number of Cars Tested	Type of Car [a]	Speed Miles per Hour	Average Carbon Monoxide in Exhaust Gas		Cubic Feet per Minute of Ventilating Air per Car Required to Dilute Carbon Monoxide to 1 Part per 10,000 (safe for indefinite period)
			Per Cent	Cubic Feet per Hour per Car (at 65 F, 29.92 in. Hg)	
Full Load and 3% Grade Upward					
7[b]	7P	3	8.4	88.5	14750
5[b]	1½T	3	9.5	65.1	10850
22	1½-3	6	6.9	114.3	19050
16	3½-4½	6	6.9	148.1	24680
18	5T	6	6.1	163.3	27220
6	7P	10	8.9	117.2	19530
7	1½T	10	7.3	86.3	14380
22	1½-3	10	6.7	149.6	24930
16	3½-4½	10	6.1	88.5	14750
18	5T	10	4.9	199.3	33220
13	7P	15	8.3	146.7	24450
12	1½T	15	6.9	117.7	19620
15	1½-3	15	5.4	130.1	21680
16	3½-4½	15	4.1	181.7	30280
Accelerating up 3% Grade with Full Load					
7[b]	7P	0 to 15	6.5	163.6	27270
5[b]	1½T	0 to 15	5.4	65.9	10980

[a] Types are designated thus: *5P* and *7P* represent 5 and 7 passenger cars, *1½T*, 1½-3, 3½-4½ and *5T* represent trucks of the respective tonnages.
[b] Tested under winter conditions.

of samples was taken at the same locations as in the last survey between 10:28 and 10:31. The analyses of these samples are tabulated in Table 1.

The curve, *gravity ventilation through fans,* Fig. 3, shows the average concentration at the 3 stations for the indicated distances above the floor. In spite of the fact that the number of cars idling was reduced by one-half, and that some ventilation was supplied, the carbon monoxide concentration in the garage had increased over the previous survey. It was visibly noticeable that the condition had not improved.

Survey 31, Gravity Ventilation Through Windows, April 16, 1932, 10:41 to 10:44 P.M.

In order to increase the effect of gravity ventilation over that shown in the last survey, 6 of the 24 windows similar to *E,* Fig. 2, were opened, 3 on the right side and 3 on the left side. This provided an open window area of 46.5 sq ft on the right side and 35.0 sq ft on the left. The air intake in the front was enlarged by opening the overhead door to 3 ft above the floor. These changes were made between 10:31 and 10:35. The 2 cars, *M* and *O,* continued idling.

The same number of samples was taken at the same stations as in the last survey between 10:41 and 10:44. The results of these analyses are tabulated

in Table 1, and the average for the 3 stations is plotted for the indicated distances from the floor in the *gravity ventilation through windows* curve, Fig. 3. That the gradient from the floor to ceiling falls off between 5 and 8 ft is probably due to the fact that the open windows in this case extended down to a point 5 ft above the floor.

The air condition in the garage had noticeably improved, particularly for the first 3 or 4 ft above the floor line. There was a noticeable stratification of visible smoke from 4 ft above the floor to the ceiling.

Survey 32, Cross Ventilation, April 16, 1932, 10:57 to 11:02 P.M.

Conditions for this survey were obtained by adjusting the 3 fans on the right hand side, *F, G,* and *H,* to blow air into, and those on the left, *I, J,* and *K,* to exhaust air from, the garage. The front door and all windows, including window *E,* in box, *L,* were closed, and the three, *M, N,* and *O,* were idling. These changes were made immediately after the last survey, and samples were taken at 3 stations and 4 levels between 10:57 and 11:02. The analyses of these samples are tabulated in Table 1. Also, the averages for the 3 stations are plotted against distance from the floor in the *cross ventilation* curve, Fig. 3. This condition of ventilation shows a consistent gradient from floor to ceiling. The air supplied by the fans on the right side of the garage fell noticeably and rather quickly to the lower regions, and probably spread out over the floor. There was some noticeable stratification of visible smoke in the upper half of the garage, excepting for a distance of about 10 ft away from the right hand wall.

Surveys 33 and 34, Downward Ventilation, April 22, 11:35 to 11:40 P.M, *11:48 to 11:52* P.M.

For these surveys all fans were arranged to blow air into the garage through the boxes at points 1 ft below the ceiling. The front door was opened 20 in. above the floor, and the window, *E,* in box, *L,* was opened; all other windows were closed. Three cars, *N, O,* and *P,* were started idling. However, one was found to have stopped sometime during the surveys.

These changes were made and the cars started idling at 11:10 P.M., and samples for Survey 33 were taken between 11:35 and 11:40. With no changes in conditions samples for Survey 34 were taken 12 min later or between 11:48 and 11:52 P.M. The results of the analyses of these samples are given in Table 1, and the averages are plotted against distance from the floor in Fig. 3, giving the *downward ventilation* curve. Little gradient from the floor to the ceiling is shown. There was practically no visible smoke in the garage and no stratification at the ceiling. There appeared to be a slight stratification of smoke in some parts of the garage midway between the floor and ceiling. The visible smoke in the exhaust coming from the rear of the idling cars continued to rise toward the ceiling, in spite of the downward ventilation, although not quite as noticeably as was the case with upward ventilation.

Surveys 35 and 36, Upward Ventilation, April 23, 12:56 to 1:14 A.M.

For these 2 surveys the fans were all adjusted so as to exhaust air from the garage through the boxes. The front door was raised 20 in. above the floor, and the window, *E,* in box *L,* was opened; all other windows were closed. These changes were made by 12:18. Three cars, *N, O,* and *P,* were kept idling.

Samples were taken at the 4 elevations above Stations *1*, *2* and *3*, between 12:56 to 1:00 A.M. Without changing conditions a second series of samples, Survey 36, was taken between 1:10 and 1:14 A.M. Analyses of these samples are given in Table 1, and the averages are plotted in the *upward ventilation* curve, Fig. 3. There was little or no visible smoke in the garage and no noticeable stratification at any level. A marked gradient in carbon monoxide concentration is shown from the floor to the ceiling. The concentration at the breathing line was 1.3 parts in 10,000, compared with 2 parts for the same condition with downward ventilation. The concentration at the ceiling was slightly higher than it was for downward ventilation.

DISCUSSION OF RESULTS OF SURVEYS

Observations made of visible smoke show that exhaust gases from an automobile rise quickly upon leaving the car, due to their higher temperature. Further, high concentrations of carbon monoxide found near the ceiling indicate that high concentrations of carbon monoxide are usually found at the ceiling. These facts would seem to prove conclusively the desirability as a general rule of removing the vitiated air at the ceiling and providing the incoming air supply at the floor.

In the studies here recorded the air supplied was at a slightly lower temperature than that prevailing in the garage, which unquestionably aids in stratification of the exhaust gases at the ceiling when this cooler air is supplied at the floor line. That this same tendency of the exhaust gases to stratify at the ceiling can be relied upon if air heated above the general room temperature is supplied at or near the floor line, does not necessarily follow. The exhaust gases must necessarily leave the car at a temperature considerably higher than the surrounding air and will therefore rise, regardless of the temperature of the air supply admitted at the floor. The only counteracting effect which the warm air supply can have must be due to its tendency to set up convection currents which may distribute the exhaust gases throughout the garage after they have risen to the ceiling.

Any ventilating system which removes the carbon monoxide from the point where it tends to collect soon after leaving the cars will obviously result in a more economical system as regards power and heating requirements for maintaining the same average concentration of carbon monoxide throughout the building or at the breathing line. Hence, it would seem desirable, wherever possible to take advantage of upward ventilation, even though it should mean supplying the air at the floor line at a slightly lower temperature even when it must be heated.

For a condition of equilibrium between carbon monoxide production and air supply in a garage the rate at which carbon monoxide is being produced and added to the air is equal to the volume of air supplied (or exhausted) times the per cent of carbon monoxide in the garage air at the point of exhaust. This calculation can be applied to the condition for *cross ventilation* and *upward ventilation* in Fig. 3.

For *cross ventilation* 3 fans each supplied 800 cfm of air, or a total of 2,400 cfm to the garage while 3 cars were running. Assuming that 6.2 or the concentration found at the ceiling represents the concentration of carbon monoxide

in the air at the exhaust openings, the amount of carbon monoxide produced by each car is equal to:

$$\frac{2400 \times 60 \times 0.00062}{3} = 29.4 \text{ cfh per car.}$$

For *upward ventilation* the 6 fans each exhausted 1,100 cfm, or a total of 6,600 cfm for the entire garage. Taking 2.4 as the concentration of carbon monoxide at the ceiling as representative of the air at the point of exhaust, the carbon monoxide produced is

$$\frac{6600 \times 60 \times 0.00024}{3} = 31.7 \text{ cfh per car.}$$

The values of 29.4 for *cross ventilation* and 31.7 for *upward ventilation* are low compared with those published by the U. S. Bureau of Mines several years ago as a result of their investigations of carbon monoxide hazards in vehicular tunnels. Table 2 is taken from the U. S. Bureau of Mines data.[7] This table indicates 37.3 cfh of carbon monoxide liberation for averages of 5 and 7 passenger cars idling, a value somewhat higher than the values computed from the results of this study. This discrepancy may be due to a difference in the condition of cars used in the two tests, a difference of car design and operation, and gasoline available between the time the U. S. Bureau of Mines survey was made about 10 years ago and the present, or it may be due to air change in the garage other than that handled by the fans.

The last column of Table 2, giving the volume of air per minute required to dilute the exhaust gases of vehicles of different types and degrees of operation to a concentration of one part of carbon monoxide in 10,000 parts of air is of value to engineers in designing the ventilating system for any garage on the basis of the cars to be handled.

The high rate at which the concentration of carbon monoxide in the air built up when cars were idling with *no ventilation* demonstrated the reason for the high prevalence of asphyxiation in single-car garages. A single car idling in a one-car garage, 10 ft $\times$ 15 ft $\times$ 8 ft, will raise the concentration of carbon monoxide in the air to 25.8 parts in 5 min, 51.5 parts in 10 min, and 103.0 parts in 20 min, assuming a rate of carbon monoxide production of 37 cfh per car. Collapse of a person at rest may result from concentrations of 25.8, 51.5, and 103.0 parts in 10,000 in 30, 20 and 12 min, respectively. If the person were active, collapse would occur sooner.

ACKNOWLEDGMENT

The authors acknowledge the kind cooperation of W. F. Pschirer of the Sheraden Garage, who made this building available for the study, and assisted in many instances to make the study a success.

DISCUSSION

W. C. RANDALL (WRITTEN): I think possibly the results recorded on carbon monoxide content are misleading if one would assume that such concentrations as

[7] Ventilation of Garages, by G. W. Jones and S. H. Katz (A. S. H. V. E. TRANSACTIONS, Vol. 29, 1923; pp. 341 to 346).

are shown in the surveys might logically be expected. In other words, two to four cars idling all the time are considerably in excess of average working conditions. In our surveys of, for instance, the Auto Service Garage, as covered in the paper Airation Studies of Garages, A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929, there was about five car hours running time per day in a garage of 500,000 cu ft, many times the size of the garage used in the investigation covered by this paper.

The information given in Fig. 3 might be misleading, as one's first conclusion would be based on the caption, that it represented a true comparison of the different methods of ventilation. I do not understand how, for instance, a greater concentration of carbon monoxide is shown for gravity ventilation through fans than with no ventilation, except that the test conditions were different. Apparently the carbon monoxide condition in test No. 29, No Ventilation, had not reached a stable condition, and continued to build up even after the fan openings and doors were opened, and two cars shut down, for test No. 30, Gravity Ventilation Through Fans. No doubt if more windows had been opened, test No. 31, the carbon monoxide would have dropped seriously to more comparable conditions as reported by Randall and Leonhard in Airation Studies of Garages.

F. C. McINTOSH (WRITTEN): It must be remembered that this paper is not presented as the result of elaborate tests, but rather as a link in a chain of evidence. Every series of tests of this sort is valuable and helps us in our quest for information.

The authors made no explanation for the higher concentration under gravity ventilation through fans than with no ventilation, in spite of the generation of CO from only two cars instead of from four. This simply indicates that the ventilation was insufficient to carry away the CO as fast as it was being generated.

In data of this sort as well as in Bridge, the "gods of distribution" are powerful. The average of a few widely varying figures is not a good guide. For instance the original readings at stations 1 and 3 show average concentrations of 6.8, 5.7, 7.3 and 6.7 from floor to ceiling or a lower amount at the ceiling than at the floor. This is probably due to both stations being near the point of generation. They are averaged, however, with only one point at a distance from the generators. Three more stations like No. 2, which was probably a better average for the garage as a whole, would give a decidedly different curve. I mention this to show that the data cannot be used for computation of total CO present. However, for the comparisons between various conditions of ventilation, stations 1 and 3 were wisely selected.

Referring to the figures for "gravity ventilation through windows," there should be some comment regarding the area of the open windows. If they are about three feet by five as the cuts would indicate, would there not be a great amount of air admitted on the side exposed to the 5 mph (440 ft per minute) breeze; much greater than computed for cross ventilation? Also, would not this type of window tend to send the air to the ceiling and stir the upper stratum? How does the author explain a comparatively high and uniformly high concentration at and above the 5 ft level under such conditions?

Assuming that the windows were well open in Survey 31, the tests show that a comparatively small amount of air, mechanically directed through the proper course, either upward or downward, gives decidedly better results than a larger amount uncontrolled, and therefore that even a garage as well exposed to the prevailing winds as this one can be helped by a mechanical ventilating system.

I am asking questions on three general points, which are not discussed in this paper, but which the authors can probably answer. The first, a purely theoretical one, is: in a space filled with two gases such as CO and air, does not the diffusion ultimately become perfect, that is, does not the CO which goes to the ceiling due to its heat and its slightly less weight, become evenly diffused throughout the space after temperatures have come to equilibrium? How rapidly will a high concentration of CO in a hot upper stratum, become diffused through the lower strata?

The second point is: what other toxic or irritating gases are generated in a garage, and what concentrations are allowable? There may be some gas such as propane (CH_4) which is as heavy as CO_2 and which, after cooling, would diffuse more rapidly toward the floor. It is unlikely that this factor need be considered on account of the high toxicity of CO but the question should be raised.

The third point is: can the diffusion of smoke consisting of floating particles of liquids and/or solids be taken as an indication of the diffusion of all the gases on which it is borne into a space? I believe that it can but shall appreciate comment on it.

J. M. DALLAVALLE (WRITTEN): The study emphasizes the value of upward ventilation by mechanical means in order to maintain a safe limit of 1 part in 10,000 of CO at all vertical points within the garage in question. The Holland Tunnel experiments and, in fact, the system of ventilation now in use bear out the effectiveness of this method in removing automobile exhaust gases. There can be no question but that this paper, coming from the Laboratory of the Society, will do much in giving a clear picture of the CO distribution in a garage with various methods of ventilation. It will serve to illustrate how quickly a state of danger can be brought about under what may appear to be a safe arrangement of open windows and presumably adequate air change. The use of the vertical CO-concentration gradient as a parameter to determine the effectiveness of ventilation is in itself a valuable contribution.

The economy of the vertical system has suggested itself to the authors and it is very significant that for the particular garage in which the experiments were conducted a little more than one air change per hour is almost adequate for the three idling cars used. Based on the fan ratings, this is approximately a 2200 cfm requirement per car. If this amount is doubled, to be well on the safe side, it is still much too low to agree with values given in the last column of Table 2. The suggestion of the investigators that this result is by virtue of the change in gasoline blends and in engine construction since the data of Table 2 were compiled, seems reasonable, but I would like to ask if a higher CO concentration would not have resulted if the cars had been permitted to idle for very long periods of time.

R. M. CONNER: The carbon monoxide problem is of great importance in the gas industry and the *A. G. A.* Laboratory was established in Cleveland primarily to study it. In seven years 20,000 different types of gas appliances have been tested and approved. It has been demonstrated that it is actually cheaper to build a safer type of appliance than it was a more hazardous one. There is a relationship to Mr. Houghten's work as I am leading up to this: Heating engineers in general should use reasonable safety factors. It was intimated in the discussion of the paper that was just read that perhaps the conclusions drawn were a trifle academic and that after all what the engineer should bear in mind is an average condition rather than the worst one. We don't do that in the gas industry. We try to see that there is a safety factor incorporated in every appliance that will be sufficient to take care of almost any condition that can be imagined, and I think that the heating and ventilating engineer ought to do the same thing in laying out his building.

Codes are being enacted all over the country and the latest one in the State of New York includes some very stringent requirements as far as carbon monoxide is concerned. With population in cities increasing from year to year, and the wider use of automobiles, I don't think we can go too far in protecting the public health from the CO hazard.

No. 935

CARBON MONOXIDE DISTRIBUTION IN RELATION TO THE VENTILATION OF AN UNDERGROUND RAMP GARAGE

By F. C. Houghten [1] (*MEMBER*) and Paul McDermott [2] (*NON-MEMBER*), Pittsburgh, Pa.

THE activities of the A. S. H. V. E. Research Laboratory have been identified with garage ventilation and its effect on removing carbon monoxide hazards since 1928 when a Technical Advisory Committee on the subject was appointed. During the fall of 1931 the Committee on Research authorized a study by the Laboratory in a small single-room, window-ventilated garage and in an underground ramp garage. Details concerning the purpose of the study and the results of the investigation in the single-room, window-ventilated, Sheraden Garage were published in another report.[3] The results of the study in the underground ramp garage in the basement of the Grant Building are the basis of this paper.

Grant Building and Garage

The Grant Building, Fig. 1, is a modern 40-story office building of the set-back skyscraper type built in 1926. A plan of the basement of the building is shown in Fig. 2. It is 48½ ft in depth below the first floor of the building, and is divided into two parts. The 5-story ramp garage in which the study was made is the west portion, 171 ft 9 in. by 91 ft 4 in. in area. The east portion of the basement, having a floor area of 5,630 sq ft, is partitioned off from the garage area by an unplastered hollow tile partition, and contains storage space and fireproof vaults for the office tenants of the building, a laundry and other service rooms, heating and other mechanical equipment for the building, and the garage ventilating plant. The division of this part of the basement into floors is somewhat irregular, but the main portion of the area is divided into 5 floors below the first floor of the building.

Fig. 3 is a typical floor plan of the garage, showing location of the ramps, the division of the floors into the east and west levels, and the vertical ventilating ducts. Fig. 4 is a vertical section of the garage, showing the floor levels,

[1] Director, Research Laboratory, American Society of Heating and Ventilating Engineers.
[2] Research Engineer, Research Laboratory, American Society of Heating and Ventilating Engineers.
[3] Carbon Monoxide Distribution in Relation to the Ventilation of a One-Floor Garage, by F. C. Houghten and Paul McDermott (A. S. H. V. E. Transactions, Vol. 38, 1932, p. 425).

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Milwaukee, Wis., June, 1932, by F. C. Houghten.

arrangement of ramps and other characteristic features. The floors are numbered from the top down as 1-east, 1-west, etc., to 5-east and 5-west. In presenting data in this report, these are frequently referred to as *1E, 1W,* etc. to *5E* and *5W* respectively. In other places these are referred to as the first and fifth floors or levels.

The ventilation of the garage includes independent supply and exhaust sys-

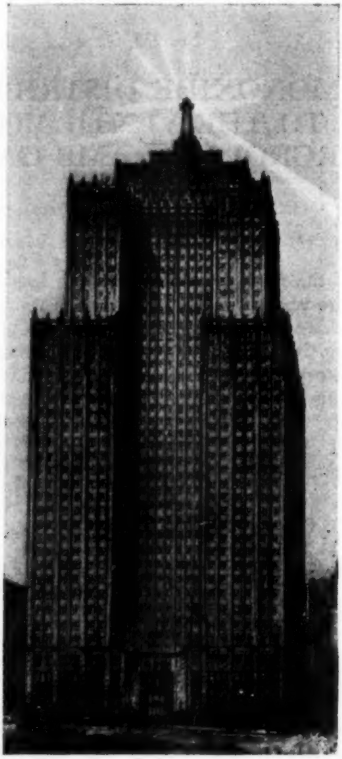


FIG. 1. GRANT BUILDING, PITTSBURGH, PENNSYLVANIA

tems. The supply side of the ventilating system consists of two intake fans located on the fifth level which take air from a 4 ft by 40 ft 8 in. concrete-lined, downcast shaft leading down from a grating in the sidewalk at the south end of the building. The incoming air passes through a 9 ft by 25 ft opening at the bottom of this shaft which is fitted with two sections of adjustable louvers, 9 ft by 7 ft and 9 ft by 18 ft, respectively. It then passes through tempering coils into a room from which it enters either of the two supply fans. The two supply fans deliver the incoming air to a common

set of concrete ducts, distributing it to the bottom of the 11 sheet metal supply risers running up to a point 18 in. below the ceiling of the first floor. These risers supply air at both ceiling and floor at the 11 points indicated in Fig. 3 on each floor level. During the heating season the supply air is tempered so that it enters the fans at approximately 55 F. The temperature in the garage is usually from 2 to 5 deg higher than that of the air entering the fans.

The exhaust system consists of 6 downcast ducts, running from near the ceiling of the first or upper floor to a system of concrete ducts below the bottom or fifth floor of the garage, from whence the exhaust air is carried to a common suction chamber from which it enters the two exhaust fans, one located on the fifth level and the other on the fourth level. Each exhaust fan discharges the vitiated air through a separate sheet-metal-lined air shaft, which runs from the fan up through the building to a setback on the thirty-fourth floor. The arrangement of the ventilating system allows great flexibility of operation. Each supply or exhaust fan may be operated independently, each functioning on all sections of the garage. All 4 fans are equipped with variable speed control.

TABLE 1. DIMENSIONS OF VERTICAL SUPPLY AND EXHAUST DUCTS

Floor	Supply Duct Dimensions								Exhaust Duct Dimensions			
	1	2 & 3	4	5	6 & 9	7 & 8	10	11	1 & 6	2	3 & 4	5
1	17x24	18x18	0	0	0	18x13	0	18x18	11x20	0	24x54	21x27
2	24x35	18x26	24x35	18x 52	24x21	18x26	18x18	18x28	18x22	21x30	33x54	21x42
3	24x51	18x39	24x51	18x 76	24x42	18x39	18x36	18x42	24x24	21x40	44x54	21x56
4	24x68	18x53	24x68	18x104	24x63	18x53	18x56	18x54	27x30	21x50	54x54	21x70
5	24x84	18x64	24x84	18x121	24x84	18x64	18x72	18x72	31x33	21x60	54x66	21x84

At low speed with the small set of adjustable intake louvers open, each of the two supply fans delivers 19,500 cfm when both are in operation, and 21,100 cfm when both are in operation with all intake louvers open. At high speed they each deliver 30,400 cfm with the one set of louvers open, and 39,600 cfm with all intake louvers open. Each of the two exhaust fans handles 12,200 cfm at low speed with both fans operating. For both fans operating at high speed they each handle 18,400 cfm.

Fig. 3 shows the location of the supply risers supplying air to the different levels of the garage, and the downcast ducts exhausting air from the different levels. The dimensions of the vertical supply and exhaust ducts are given for each floor in Table 1. The outlet and inlet sizes and their locations for each floor are given in Table 2.

The garage, which has a parking capacity of 237 cars, is of the usual ramp type, the floors on the east and on the west half of the garage being staggered by one-half story. Each floor, east or west, provides a roadway in the center, with angle parking at either side. Moving cars, ascending or descending, use this roadway. At either end of the garage, ramps lead from any roadway or floor level to the floors above and below. All cars, either ascending or descending the ramps, pass northward in the east half of the garage and southward in the west half.

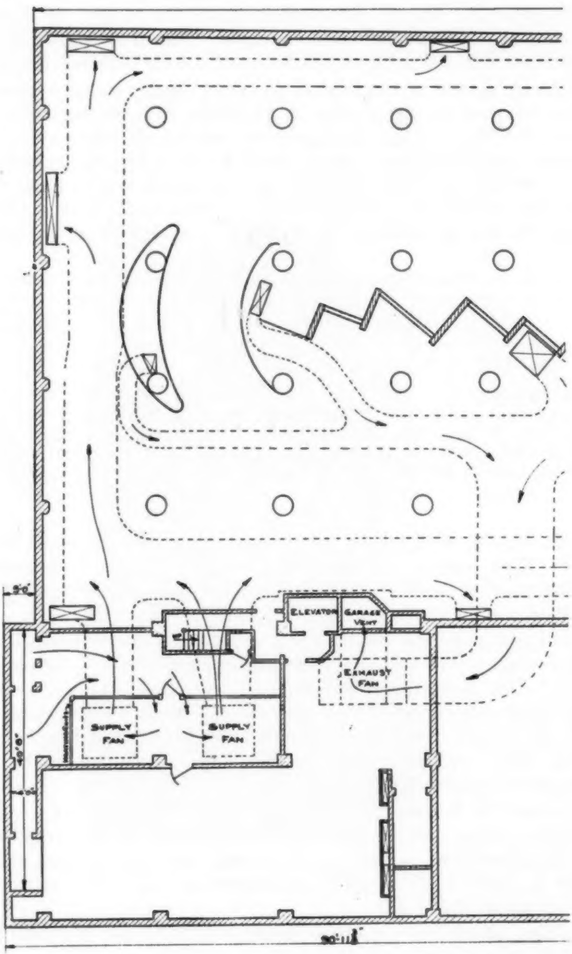
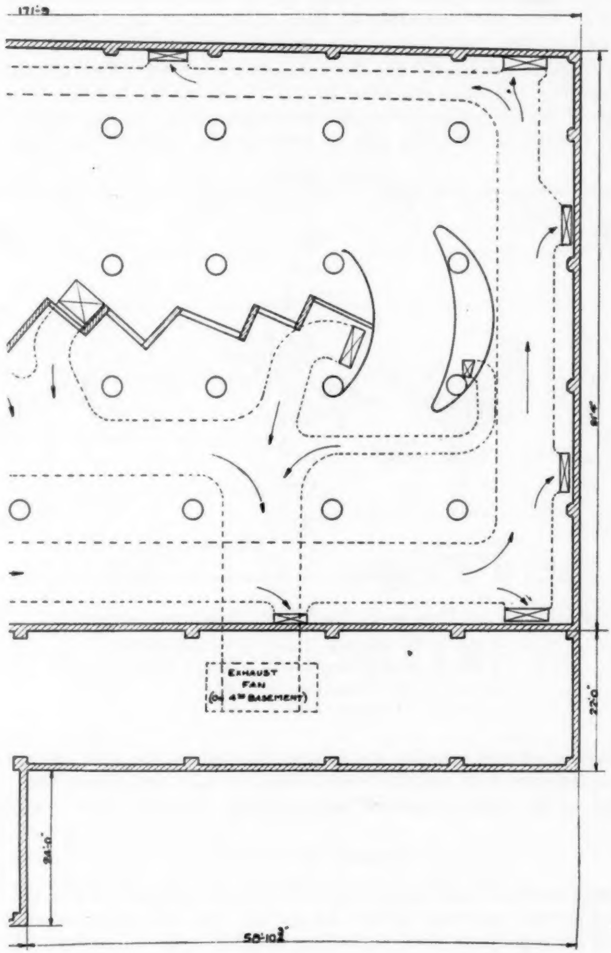


FIG. 2. GRANT BUILDING



Garage—Basement Plan

All floors are interconnected through the ramps and throughout the center line of the garage where the staggered floors meet. There are also four vertical openings throughout the height of the garage, two near the ramps at either end. This interconnection of floors allows free air movement throughout the garage, thus complicating any study of the ventilating effects.

The garage is essentially for the convenience of the office tenants of the Grant Building, but when not filled by the tenants of the building it is opened to the public. In normal operation, the maximum number of cars enters between 8:30 and 10:00 A.M. when the rate of entering cars reaches 2 to 3 per minute.

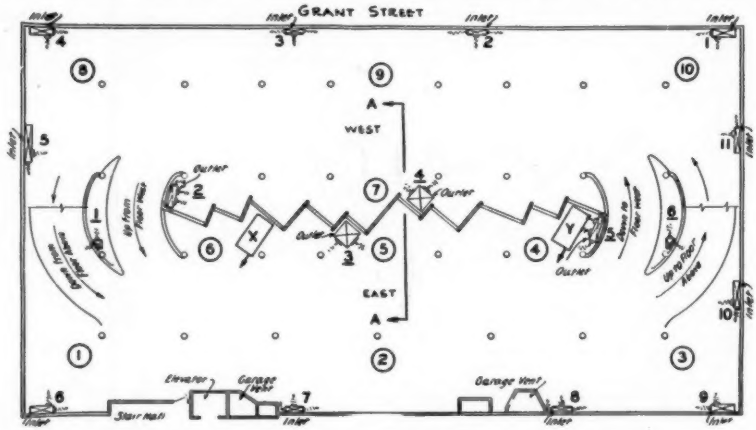


FIG. 3. TYPICAL FLOOR—GRANT BUILDING GARAGE

1 etc. to 11—Supply Risers (Around Wall)
<u>1</u> etc. to <u>6</u>—Exhaust Ducts (Center Line)
① etc. to ⑩—Sampling Stations
X and *Y*—Location Idling Cars

The movement of cars remains at minimum throughout the day, and again rises to a peak between 4:30 and 6:00 P.M. when the cars are driven out. During the period of this study the garage was operated at about 75 per cent capacity.

METHOD OF ATTACK

The study resolved itself into a series of surveys designed to give the concentration of carbon monoxide within the garage, and the variation in this concentration from point to point, depending upon the type and amount of ventilation and the car operation.

Two methods of determining the concentration of carbon monoxide in the air were available for this study, namely, the pyrotannic acid method described in the companion report [3] and the Bureau of Mines Carbon Monoxide Recorder, which is sensitive to concentrations of one part per million. The

[3] Carbon Monoxide Distribution in Relation to the Ventilation of a One-Floor Garage, by F. C. Houghten and Paul McDermott (A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932, p. 425).

recorder, a complicated instrument developed by the U. S. Bureau of Mines as reported by S. H. Katz and others,[4] Fig. 5, is not easily moved, and hence, in making a garage survey with it, the instrument must be set up at a convenient place, and samples from the various points piped to it.

A considerable time lag is experienced in the use of the recorder, making it necessary to sample a given location for at least 10 min in order to get a true indication of the carbon monoxide concentration. In making the study in the Grant Building Garage, the recorder was located on the third floor east near the center of the building. A ¾-in. pipe was run from the instrument throughout the height of the garage with valved outlets at each floor east and west. A rubber hose could be attached at any one of these outlets, and its nozzle carried to any point on that floor in making the survey. While this method of analysis has the advantage of being extremely sensitive, it also

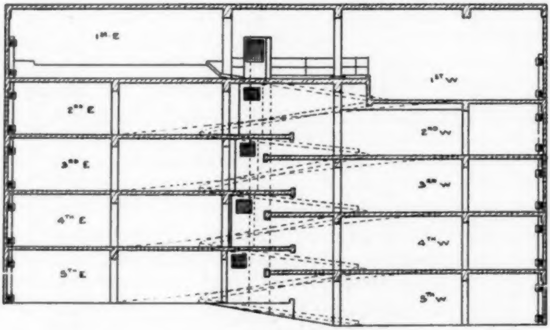


FIG. 4. VERTICAL SECTION *AA* OF GRANT BUILDING GARAGE SHOWING FLOORS AND RAMPS

has the disadvantage of not being adaptable to a rapid survey of an entire floor. A combination of the use of the recorder and the pyrotannic acid method was found convenient and valuable. The carbon monoxide recorder was used as a control to determine variation in concentration at one point with time, or to determine variations at a few points. The wider surveys of the garage were made by taking samples for the pyrotannic acid method of analysis when the recorder showed that the desired condition existed.

SURVEYS MADE IN THE GARAGE

Eighteen surveys were made in the garage, representing different conditions of ventilation of car operation. The conditions for the separate surveys are presented, together with the results of the analyses and conclusions drawn from them.

[4] *U. S. Bureau of Mines Technical Paper*, 355, by Katz, Reynolds, Frevert and Bloomfield (1926).

Survey 2 of Entire Garage by Recorder Method, March 22, 1932, 10:57 A.M. *to 5:11* P.M.

For this survey the supply and exhaust fans were operated at low speed throughout the day, while the recorder sampled air from 12 stations throughout the garage. At each station samples were taken at both ceiling and floor. Sampling at some of the stations was repeated. As stated the recorder shows

TABLE 2. OUTLET AND INLET SIZES AND THEIR LOCATIONS FOR EACH FLOOR

Supply Duct Number	Floor 1			Floors 2, 3, 4, 5		
	Number of Outlets	Height and Width of Outlets in Inches	Height Floor to Center	Number of Outlets	Height and Width of Outlets in Inches	Height Floor to Center
1	2 2	13 x 14 13 x 14	90 12	2 2	12 x 13 12 x 13	86 12
2, 3, 7, 8*	3 3	8 x 10 8 x 10	82 10	3 3	8 x 10 8 x 10	88 10
4				2 2	12 x 13 12 x 13	86 12
5	2	14 x 24	30	2 2	12 x 20 12 x 20	90 12
6 and 9				2 2	12 x 18 12 x 18	90 12
10				2 2	12 x 14 12 x 14	90 15
11	2 2	10 x 12 10 x 12	101 6	2 2	10 x 12 10 x 12	77 11

Exhaust Duct Number	Floor 1			Floors 2, 3, 4, 5		
	Number of Outlets	Height and Width of Outlets in Inches	Height Floor to Center	Number of Outlets	Height and Width of Outlets in Inches	Height Floor to Center
1 and 6*	1 1	17 x 20 12 x 18	80 78	1 1	10 x 18 10 x 10	89 89
2				2 2	12 x 18 12 x 18	134 24
3 and 4	1 1	48 x 48 48 x 24	60 60	1 1	24 x 48 48 x 24	84 72
5	2 1	36 x 13 36 x 25	66 66	2 1	26 x 18 13 x 18	75 75

* Supply duct No. 8 has an extra outlet 8 x 10 in. at 60 in. from floor on 4th floor. Exhaust duct No. 1 has an extra outlet 10 x 18 in. at 89 in. from floor on 2nd floor.

considerable lag in indicating carbon monoxide concentration for any given air condition. Sampling for a period of 10 min gave at least 7 min during which the instrument record showed the true percentage of carbon monoxide at the point studied. The carbon monoxide concentrations found for all stations in this survey throughout the day are shown in Table 3.

The station numbers and floors used in this table are indicated in Figs. 3 and 4 respectively. The time at which sampling at any station was started is indicated in the first of the three columns under the station; the second and

third columns under the stations give the concentrations found at the ceiling and floor of the station, the ceiling being sampled first.

This survey shows concentrations throughout the garage varying from 0.113 to 0.315 parts of carbon monoxide per 10,000 parts of air. During the day studied frequent observations were made of the carbon monoxide concentration at the single control point, 5. Fig. 6 is a log of the recorder reading during the day while sampling at the various locations indicated and at the control point. The evening peak is shown between 5:00 and 6:30 P.M. All of this peak is not included in the foregoing survey.

Little or no consistent variation in carbon monoxide concentration throughout the garage is shown, excepting that due to the time of day at which the analysis was made. There is, however, a fairly consistent higher percentage shown at the ceiling, the average of all ceiling observations at all stations during the entire day giving a concentration of 0.188 parts in 10,000, and the floor an average of 0.178 parts in 10,000. Eighty-eight cars passed through, entered, or left the floor on which samples were taken during the period of the survey, or an average of 14.6 per hour. Bearing in mind that concentrations of 1.0 part in 10,000 may be considered permissible over an extended period of time, and that concentrations up to 2.0 parts in 10,000 are not serious for short intervals, it will be noted that even with the ventilating system running at low speed the condition is well below the permissible value of 1.0 part in 10,000, excepting for a short interval in the evening rush period, when it went above 1.0 part for 40 min, reaching a maximum of 1.3 parts in 10,000 for a much shorter time.

In some of the later tests steps were taken to increase the concentration in order to obtain data on conditions of higher concentration. In considering these later cases, it should be kept distinctly in mind that they were artificial conditions set up for the purpose of the study, and were by no means characteristic of the garage.

Survey 3 of Floor 3E by Recorder Method, March 23, 1932, 10:23 A.M. *to 5:32* P.M.

This survey was limited to Floor *3E*. Both supply and exhaust fans were operated at low speed. All of the air supply openings at the ceiling and all the exhaust openings at the floor on *3E* were closed, allowing air to be supplied at the floor line and to be exhausted at the ceiling.

Throughout the day samples were taken at 5 stations, *1*, *2*, *3*, *4*, and *7*. The results of these analyses are given in Table 3. The average for all samples taken at the ceiling in this survey gives a concentration of 0.246; the average for the floor is 0.206. Eighty cars passed through, entered, or left this floor during the survey period, or an average of 11.4 per hour.

Survey 4 of Floor 3E by Recorder Method, March 24, 1932, 10:00 A.M. *to 5:10* P.M.

This survey is similar to Survey 3, excepting that the direction of the ventilation on Floor *3E* was reversed; i.e., the air supply openings were closed at the floor and the exhaust openings were closed at the ceiling. In this survey the average of all analyses showed a concentration of 0.288 at the ceiling and 0.229 at the floor. Ninety-five cars passed through, entered, or left this floor during the survey period, or an average of 13.5 per hour.

Survey 5 of Stations 1 and 3, Floors 2E, 3E, and 5E, by Recorder Method, March 25, 1932, 8:33 A.M. *to 5:45* P.M.

Both supply and exhaust fans were run at low speed. Riser 6, the air supply riser in the southeast corner of the garage (Fig. 3), was entirely closed on all floors, so that this corner of the building did not receive an air supply from any point closer than Riser 7, 44 ft north along the east wall, and Riser 5, 45½ ft west along the south wall. The purpose of this test was to demonstrate

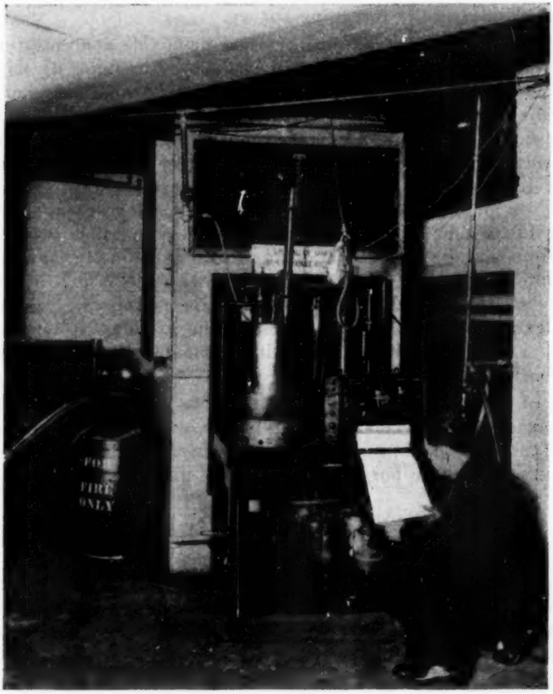


FIG. 5. U. S. BUREAU OF MINES CARBON MONOXIDE RECORDER SET UP IN GRANT BUILDING GARAGE

whether or not such an unventilated corner resulted in the building up of a high concentration of carbon monoxide.

Samples were taken at the ceiling and floor at Stations 1 and 3 on Floors *2E, 3E,* and *5E*. The results of the individual analyses are given in Table 3, together with averages for ceiling and floor for each station and floor number, and averages for ceiling and floor for each station on all three floors, and a final average for both ceiling and floor at each station represented by the two corners of the garage on all floors studied. These averages show concentrations of 0.488, 0.254, and 0.268 at the ceiling, and 0.435, 0.240, and 0.284

at the floor for Station 1 on *2E, 3E,* and *5E* respectively, giving an average of 0.320 for the ceiling and 0.301 for the floor for all samples taken at Station 1, or in the southeast corner, and an average of 0.311 for both ceiling and floor in this corner of the building. In the northeast corner, where the ventilation was not shut off, averages of 0.330, 0.189, and 0.284 were found at the ceiling, and 0.450, 0.144, and 0.196 at the floor line of the three floors respectively, giving averages of 0.250 and 0.246 at the ceiling and floor for all 3 floors in this corner, and a total average of 0.248 for both ceiling and floor of all levels in this corner of the building. One hundred ninety-five cars passed through, entered, or left the floors on which samples were taken during the period of the survey, or an average of 20.2 cars per hour.

These averages indicate a higher concentration for both stations at the higher floors. They also show a fairly consistent, though not universal, higher concentration at the ceiling. The averages also show a higher concentration for both ceiling and floor for the southeast corner in which the air supply was shut off.

Survey 9 of Floors 1W, 3E, and 4E, by Recorder Method, March 28, 1932, 10:03 A.M. *to 6:00* P.M.

In this survey air supply Risers 1, 2, 8, 9, 10, and 11 (Fig. 3), or all of the air supply risers in the north end of the building, were closed on all floors. Likewise, downcast exhaust Ducts 1, 2, and 3 or all of the exhaust ducts in the south end of the building were closed on all floors, thus allowing air to be supplied only in the south half of all floors, and exhausted in the north half. Samples were analyzed on the recorder throughout the day at Stations 1, 3, 5, 7, 8, and 10 on Floors *1W, 3E,* and *4E.* One hundred forty-one cars passed through, entered, or left the floors on which samples were being taken during the period of the survey, or an average of 17.6 cars per hour.

The average analyses of all these samples which are shown in Table 3 are as follows: Stations 1 and 8, the south or air supply end, 0.183; Stations 3 and 10, or the extreme north end, 0.250; Stations 5 and 7 in the center of the garage near the main exhaust duct which was in operation at the time of the survey, 0.277; showing the lowest concentration near the air supply, as would be expected. However, the highest concentration was not found in the dead corners, Stations 3 and 10, where it might have been expected, but rather near the exhaust openings.

Survey 22 of Floor 3E by the Recorder Method, March 30, 1932, 1:44 P.M. *to 6:05* P.M.

This survey was made on the third floor east, with the supply and exhaust fans running at low speed. Supply Riser 6, Fig. 3, in the southeast corner was blanked off on this floor. All other air supply vents were open.

One car stationed as indicated by *X* in Fig. 3 was idling throughout the survey. Samples were taken at the ceiling and floor of Stations 1, 2, and 5. The average during the survey shows concentrations of 0.462 and 0.515 at the floor and ceiling respectively of Station 1, 0.788 and 1.449 at the floor and ceiling of Station 2, and 0.995 and 1.347 at the floor and ceiling of Station 5. This survey shows marked increase in concentration at the ceiling over that of the floor, but does not show a higher concentration in the unventilated corner.

TABLE 3. CARBON MONOXIDE CONCENTRATIONS AT FLOORS AND STATIONS FOR EACH FLOOR—RECORDER

Survey No.	Date	Floor	Stations 1 Time	1 C.	1 F.	2 Time	2 C.	2 F.	3 Time	3 C.	3 F.	4 Time	4 C.	4 F.	5 Time	5 C.	5 F.
2	Mar. 22	2E							12:59 P.M.	.170	.113				12:32 P.M.	.202	.221
2	"	4E							2:54 P.M.	.126	.176				2:28 P.M.	.252	.158
2	"	5E							4:01 P.M.	.139	.107				3:32 P.M.	.315	.221
3	Mar. 23	3E	10:56 A.M.	.126	.082	11:19 A.M.	.095	.126	12:05 P.M.	.082	.158	11:41 A.M.	.221	.101			
3	"	3E				4:25 P.M.	.252		5:11 P.M.	.504	.567						
4	Mar. 24	3E	10:25 A.M.	.201	.170	10:47 P.M.	.176	.315	11:34 A.M.	.239	.126	11:14 A.M.	.302	.177			
4	"	3E	4:25 P.M.	.284	.176	4:05 P.M.	.252	.201	4:49 P.M.	.378	.391						
5	Mar. 25	2E	4:05 P.M.	.202	.221				3:45 P.M.	.139	.196						
5	"	2E	4:45 P.M.	.328	.328				4:25 P.M.	.284	.365						
5	"	2E	5:25 P.M.	.933	.756				5:05 P.M.	.567	.788						
5	Averages	2E		.488	.435					.330	.450						
				.462						.390							
5	Mar. 25	3E	8:33 A.M.	.315	.346				8:58 A.M.	.441	.252						
5	"	3E	9:19 A.M	.410	.315				9:39 A.M.	.189	.170						
5	"	3E	9:59 A.M.	.504	.504				1:45 P.M.	.095	.013						
5	"	3E	1:25 P.M.	.095	.126				2:25 P.M.	.170	.158						
5	"	3E	2:05 P.M.	.095	.032				3:05 P.M.	.050	.126						
5	"	3E	2:45 P.M.	.107	.114												
5	Averages	3E		.254	.240					.189	.144						
				.247						.167							
5	Mar. 25	5E							11:10 A.M.	.378	.328						
5	"	5E	11:30 A.M.	.410	.378				11:50 A.M.	.189	.063						
5	"	5E	12:10 P.M.	.126	.189												
5	Averages	5E		.268	.284					.284	.196						
				.276						.240							
5	Averages	2E, 3E, 5E		.320	.301					.250	.246						
				.311						.248							
9	Mar. 28	4E	3:31 P.M.	.158	.095				2:50 P.M.	.107	.076				3:10 P.M.	.195	.177
9	Averages	1W, 4E		.127						.092						.186	

TABLE 3 (CONCLUDED). CARBON MONOXIDE CONCENTRATIONS AT FLOORS AND STATIONS FOR EACH FLOOR—RECORDER

SURVEY No.	DATE	FLOOR	STATIONS 6 TIME	6 C.	6 F.	7 TIME	7 C.	7 F.	8 TIME	8 C.	8 F.	9 TIME	9 C.	9 F.	10 TIME	10 C.	10 F.
9	Mar. 28	3E	10:43 A.M.	.126	.076				10:03 A.M.	.221	.189				10:23 A.M.	.195	.195
9	"	3E	5:15 P.M.	.284	.346				11:04 A.M.	.315	.252				4.51 P.M.	.441	.441
9	"	3E							4:04 P.M.	.378	.239						
9	"	3E							5:38 P.M.	.806	.756						
9	Averages	3E		.208						.394						.318	
22	Mar. 30	3E	1:44 P.M.	.504	.347	2:05 P.M.	1.449	.756							1:24 P.M.	1.512	.819
22	"	3E	2:45 P.M.	.347	.284	3:08 P.M.	1.701	1.071							2:25 P.M.	1.197	.882
22	"	3E	4:05 P.M.	.693	.756	4:15 P.M.	1.197	.536							3:31 P.M.	1.827	.882
22	"	3E													4:35 P.M.	1.323	.914
22	"	3E													4:55 P.M.	1.323	1.323
22	"	3E													5:15 P.M.	1.323	1.134
22	"	3E													5:35 P.M.	1.260	1.008
22	"	3E													5:55 P.M.	1.008	
22	Averages	3E		.515	.462		1.449	.788								1.347	.995
25	Apr. 1	3E	12:35 P.M.	1.575	.945				1:20 P.M.	1.260	.315				1:40 P.M.	1.070	.788
25	"	3E	2:12 P.M.	2.140	.334				2:57 P.M.	.945	.662				3:27 P.M.	1.200	1.200
25	"	3E	3:47 P.M.	1.480	.472				4:47 P.M.	1.389	1.480				5:07 P.M.	1.260	.945
25	"	3E	5:27 P.M.	1.575	.694				6:07 P.M.	2.395	.882				6:27 P.M.	1.070	.567
25	Averages	3E		1.693	.611					1.498	.835					1.115	.875
28	Apr. 2	3E	10:34 A.M.	.693	.473	10:56 A.M.	.378	.211									
28	"	3E	11:16 A.M.	.466	.473	11:36 A.M.	1.830	.504									
28	"	3E	11:56 A.M.	.819	.513	12:16 P.M.	1.890	.882									
28	"	3E	12:36 P.M.	.756	.567	12:56 P.M.	2.394	.756									
28	"	3E	1:16 P.M.	.504	.630												
28	Averages	3E		.647	.531		1.623	.588									
2	Mar. 22	1W							12:05 P.M.	.189	.252	11:27 A.M.	.113	.202			
2	"	3W				1:35 P.M.	.176	.221	1:59 P.M.	.265	.139						
2	"	5W				4:36 P.M.	.095	.076	5:01 P.M.	.126	.158						
3	Mar. 23	3E	10:33 A.M.	.441	.202												
4	Mar. 24	3E	10:05 A.M.	.473	.273												
9	Mar. 28	1W				1:05 P.M.	.132	.453	1:22 P.M.	.284	.189				12:45 P.M.	.258	.252
9	"	1W				2:01 P.M.	.315	.410	2:21 P.M.	.221	.158				1:41 P.M.	.284	.252
9	Averages	1W, 4E					.328			.213						.263	

Survey 25 of Floor 3E by Recorder Method, April 1, 1932, 12:35 P.M. *to 6:47* P.M.

During this survey both supply fans were run at low speed. Only one exhaust fan was run at low speed. In other respects the ventilation was normal. Two cars were allowed to idle on Floor *3E* throughout the test, one as indicated by *X* and the other as indicated by *Y*, Fig. 3. Samples were taken at Stations 1, 3, and 5, during the period of the survey with the following average concentrations: Station 1, 0.611 and 1.693 at the floor and ceiling respectively; Station 3, 0.835 and 1.498 at the floor and ceiling respectively; Station 5, 0.875 and 1.150 at floor and ceiling respectively. The average for all 3 stations gave 0.774 at the floor and 1.447 at the ceiling.

Survey 28 of Floor 3E by Recorder Method, April 2, 1932, 10:12 A.M. *to 1:36* P.M.

Both supply and exhaust fans were run at low speed. All air supply and exhaust vents on Floor *3E* were closed allowing no direct ventilation on this floor. One car was idling as indicated by *X*, Fig. 3. Ceiling and floor samples were analyzed at Stations 1 and 2. Average carbon monoxide concentrations of 1.135 and 0.559 were found at the ceiling and floor, respectively.

It was considered quite surprising that this condition, with no direct ventilation on the floor, resulted in a lower concentration of carbon monoxide at both ceiling and floor than Survey 22, which was made for the same conditions with ventilation. A survey of air movement as indicated by smoke drift showed a marked air change on this level with no ventilation, the air drifting in through the lower half of the elevation of the space from *3W* below, and out through the upper half of the floor space into *2W* above.

Survey 7 of Floors 1W, 2E, 3E, 3W, 5E, and 5W by the Pyrotannic Acid Method, March 25, 1932, 9:28 to 9:38 P.M.

This survey was made by the pyrotannic acid method of analysis. Both supply and exhaust fans were running at low speed. The ventilating system throughout the garage was otherwise running normally. Two idling cars were placed, one near either corner of each of the floors, east and west, making twenty idling cars in the entire garage; in addition, a car occasionally passed in or out.

The survey was made by taking samples of air in the bottles at Stations 1, 2, 3, 5, 7, 9, and 10 on Floors *1W, 2E, 3E, 3W, 5E,* and *5W*. Six observers did the sampling, resulting in all samples being taken within a 10-min period from 9:28 to 9:38 P.M.

The concentrations shown by the samples at the different stations are given in Table 4. The analysis shows little consistent variation as between ceiling and floor. However, it does show a progressive rise in concentration from the lowest floor to the highest, with the exception of *3E*. At the close of the test, it was found that one of the two cars idling on this floor had stopped.

The condition of the test was obtained by starting all cars idling with the ventilation shut down. Fig. 7 shows how the concentration of carbon monoxide as indicated by the recorder increased between the time the cars were started at 8:00 to 8:10 P.M. and the time when the ventilation was started up at full speed at 8:59 P.M. The fans running at full speed brought the

concentration of carbon monoxide down rapidly from the maximum of something over 7.5 parts in 10,000 to 3.2 parts in a period of 15 min. The fans were again stopped at this point and started up at low speed at 9:20. At 9:28 P.M. the concentration was found to be fairly consistent, or in other words, the rate of carbon monoxide generation, together with the air change, approached equilibrium.

Survey 10 of Floor 3E and 3W by Pyrotannic Acid Method, March 28, 1932, 9:35 P.M. to 9:55 P.M. with Moving Cars.

This survey was made with the pyrotannic acid method. The desired condition in the garage was obtained by driving five cars up and down the ramps from Floor *4E*, or one floor below where the study was made, up to *2E*, or one floor above the floor on which the study was made, and back down. Thus, in making the circuit, each car passed through *3E* twice. This arrangement gave about seven cars passing through this floor per minute. The U. S. Bureau of Mines Carbon Monoxide Recorder was kept in operation sampling the air on Floor *3E* as a control.

Both supply and exhaust fans were run at low speed. All exhaust ducts in the south end of the building, or Ducts 1, 2, and 3, were closed while all supply risers in the north end of the building or Ducts 1, 2, 8, 9, 10, and 11, were closed, allowing air supply at the south end of the building only, and exhaust at the north end. Stations 1, 2, 3, 5, 7, 8, 9, and 10, on Floors *3E* and *3W* were sampled between 9:35 and 9:55. The results of the analyses are given in Table 4.

Stations 1 and 8, 3 and 10, and 5 and 7 in the south end, north end, and center of the garage show average concentrations of 1.55 and 1.90, 2.20 and 2.45, and 2.55 and 2.85 at the ceiling and floor of the south end, north end, and the center of the garage, respectively. The average for both ceiling and floor for the south end, north end, and the center of the garage shows 1.73, 2.33 and 2.70, respectively.

Survey 11 of Floor 3E by Pyrotannic Acid Method, March 28, 1932, 10:08 to 10:17 P.M.

The condition of ventilation in the garage was the same as in Survey 10, with the exception that on the third floor east the air supply was cut off at the floor and the exhaust was cut off at the ceiling, allowing air to be supplied for this space at the south end half of the garage at the ceiling, and allowing it to be exhausted at the floor through Ducts 4, 5, and 6, at the north end of the garage. Samples were taken of air for pyrotannic acid analysis at five stations on Floor *3E* between 10:08 and 10:17 P.M. These results are given in Table 4. No consistent variation in carbon monoxide concentration with station or elevation is noticeable.

Survey 12 of Floor 3E by Pyrotannic Acid Method, March 28, 1932, 10:25 to 10:31 P.M.

The condition of ventilation for this survey was the same as the condition for Surveys 10 and 11, with the exception that in this survey all air supply on the third floor east was shut off at the ceiling line, and all exhaust was shut off at the floor line, allowing air to be supplied to the third floor east at the floor line only from those supply risers located in the south half of the garage, or Risers 3, 4, 5, 6, and 7, and to be exhausted at the ceiling line

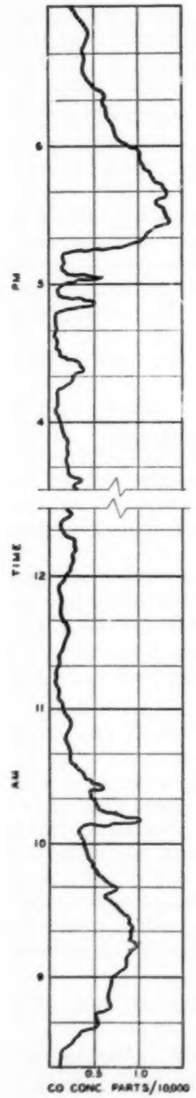


FIG. 6. (LEFT) LOG FROM RECORDER RECORD MARCH 22, 1932. FANS RUNNING SLOW SPEED. OTHER OPERATING CONDITIONS NORMAL

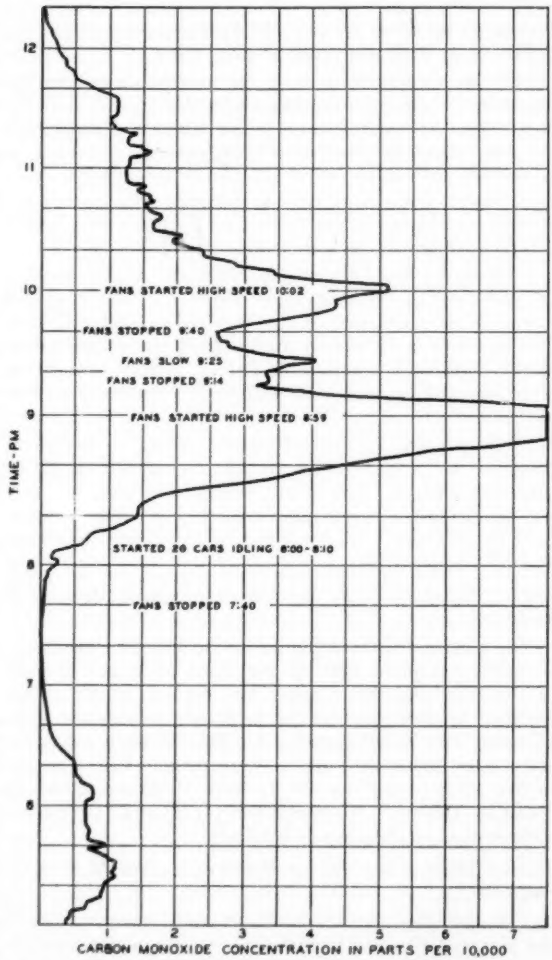


FIG. 7. LOG FROM RECORDER RECORD MARCH 25, 1932. TWENTY CARS IDLING. ABNORMAL VENTILATION AS INDICATED

only through Ducts 4, 5, and 6. Samples for pyrotannic acid analysis were taken at the ceiling, breathing line and floor of Stations 1, 2, 3, and 5, on Floor *3E* between 10:25 and 10:31 P.M. The concentrations shown by the analyses of this survey are given in Table 4. No consistent variation in carbon monoxide concentration with station or elevation is noticeable.

Survey 18 of Floors 3E and 3W by Pyrotannic Acid Method, March 29, 1932, 9:10 to 9:20 P.M.

For this survey both supply and exhaust fans were running at low speed. Otherwise, the ventilating system was running normally. Six cars were driven on Floors *2E* to *4E,* as in Surveys 10, 11, and 12, resulting in seven cars per minute passing through Floor *3E*. Samples were taken at the floor, breathing line, and ceiling of stations 1, 2, 3, 5, 7, 8, 9, and 10 on Floors *3E* and *3W*. The results of the analyses are given in Table 4. No consistent variation in carbon monoxide concentration with station or elevation is noticeable.

One sample at the breathing line at Station 5 on Floors *1E, 2E, 4E,* and *5E* and at Station 7 on Floors *1W, 2W,* and *5W* were also taken to indicate the effect of the cars running on Floors *2E* to *4E* on the floors above and below. These results are also given in Table 4, and show concentrations of carbon monoxide at the breathing line of the top floor practically as high as were found on Floor *3E*. There is a slight falling off with descent into the garage until the center of the car activity was reached. Below *4E,* the lowest floor on which cars were operating, the concentration of carbon monoxide was negligible.

Survey 19 of Floor 3E by Pyrotannic Acid Method, March 29, 1932, 9:35 to 9:44 P.M.

Both supply and exhaust fans were operated at low speed. All air supply outlets on Floor *3E* were closed at the floor, and all exhaust outlets were closed at the ceiling, admitting air only at the ceiling and removing it at the floor. The six cars were operated as in Survey 10. When conditions had reached equilibrium as indicated by the U. S. Bureau of Mines Carbon Monoxide Recorder, samples were taken at the ceiling, breathing line and floor at Stations 1, 2, 3, and 5, on Floor *3E*. Samples were also taken at the breathing line of Floors *1E, 1W, 2E, 4E, 5E,* and *5W*. No consistent variation in carbon monoxide concentration with station or elevation is noticeable on *3E*. The same variation in carbon monoxide concentration between the breathing line of *1E* and that of *5W* is evident as was shown in Survey 18.

Survey 20 of Floor 3E by Pyrotannic Acid Method, March 29, 1932, 10:11 to 10:19 P.M.

Both supply and exhaust fans were operated at low speed. All supply openings on Floor *3E* were closed at the ceiling, and all exhaust openings were closed at the floor line, thus admitting air by direct ventilation only at the floor line and exhausting it at the ceiling. The 6 cars were operated between Floors *2E* and *4E,* as in Survey 10. The U. S. Bureau of Mines Carbon Monoxide Recorder was operated as a control, and when the carbon monoxide concentration had reached equilibrium samples were taken at the ceiling, breathing line, and floor at four stations on *3E*. The results of these analyses

TABLE 4. CARBON MONOXIDE CONCENTRATIONS AT FLOORS AND STATIONS FOR EACH FLOOR—PYROTANNIC

Survey No.	Date	Time	Floor	Stations																							
				1			2			3			5			7			8			9			10		
				F.	B.L.	C.	F.	B.L.	C.	F.	B.L.	C.	F.	B.L.	C.	F.	B.L.	C.	F.	B.L.	C.	F.	B.L.	C.	F.	B.L.	C.
7	Mar. 25	9:20– 9:31 P.M.	1E & 1W						5.7							4.7		4.7				3.2		3.2	3.9		3.9
7	"	"	"																			3.2		2.5	4.7		3.2
7	"	"	2E	3.2		2.5	3.9		5.7				2.5		2.5												
7	"	"	"	2.5		2.5	3.2		3.2																		
7	"	"	3E & 3W				2.5		3.0	2.5		3.2	1.9		0.8	3.2		3.2	1.9		3.2	2.5		2.5			
7	"	"	"				1.9		1.9	2.5									3.0		3.2	2.5		2.5			
7	"	"	5E & 5W				2.5		2.5	2.5		2.5	2.5		2.5	3.2		1.9	1.9		1.9	0.8		2.5			
7	"	"	"				2.5		3.2	2.5		1.3							3.2		2.5	3.2		2.5			
10	Mar. 28	9:35– 9:55 P.M.	3E & 3W	2.5	1.9	1.3	1.9	1.6	3.2	3.0	2.5	2.5	3.2	2.5	3.2	2.5	1.9	1.9	1.3	1.9	1.8	1.8	2.0	1.3	1.9	2.1	1.9
11	"	10:08–10:17 P.M.	3E	1.9	3.2	1.9	2.5	1.9	2.5	1.9	3.2	1.8	3.2	2.1	1.9												
11	"	"	"				2.5	3.2	2.5																		
12	"	10:25–10:31 P.M.	3E	2.5	3.2	2.5	2.3	3.2	3.2	3.2	2.5	1.9	2.5	1.9	2.1												
12	"	"	"				3.2	2.5	3.2																		
18	Mar. 29	9:10– 9:20 P.M.	3E & 3W	2.5	3.2	2.5	2.5	2.5	2.5	3.2	3.2	2.5	2.5	3.2	3.2	2.5	3.8	3.8	3.8	3.8	2.5	2.1	2.5	2.1	1.9	3.2	3.2
18	"	"	"				3.6	3.2	3.2													3.0	3.8	3.8			
18	"	"	1E & 1W											2.5			1.3										
18	"	"	2E & 2W											1.9			3.2										
18	"	"	4E											3.0													
18	"	"	5E & 5W											0.3			0.3										
19	"	9:35– 9:44 P.M.	3E	3.2	3.2	3.2	3.4	3.8	3.2	2.5	3.0	3.4	3.2	3.2	3.4												
19	"	"	"				2.5	3.4	3.4																		
19	"	"	1E & 1W											3.2			3.2										
19	"	"	2E											2.9													
19	"	"	4E											3.2													
19	"	"	5E & 5W											0.3			0.3										
20	"	10:11–10:19 P.M.	3E	3.2	3.2	3.2	3.2	3.8	3.8	3.8	3.2	3.8	3.8	3.4	3.4												
20	"	"	"				2.5	3.2	3.8																		
26	April 1	3:00– 3:15 P.M.	3E	0.3	1.7	1.3				1.0	1.3	1.3	0.7	1.3	1.3												
27	April 2	12:20–12:29 A.M.	3E	0.8	0.8	0.8				0.8	0.8	1.3	0.8	0.8	1.3												

are given in Table 4. No consistent variation in carbon monoxide concentration with station or elevation is noticeable.

Survey 21 Behind an Idling Car by Pyrotannic Acid Method, March 29, 1932, 11:30 to 11:45 P.M.

With both supply and exhaust fans operating at low speed, and the ventilating system otherwise operating normally, a survey was made behind a stationary, idling car. Twenty-five samples were taken for analysis, and the results of these analyses are charted with respect to distance and elevation from the rear of the car in Fig. 8. Most of the samples were taken in a vertical plane passing through the exhaust of the car. Three samples were taken at a distance of 16 ft back of the car and 8 ft to the right, and another 3 samples the same distance back of the car and 8 ft to the left. Samples were also taken 9 ft back of the car and 8 ft to the left. There was a visibly noticeable slight drift of the exhaust gas from right to left which carried it over towards these last mentioned points of sampling.

The analyses show a concentration of 20+ parts of carbon monoxide in 10,000 about 6 in. back of the exhaust opening of the car and directly in the path of the hot gases. The survey further shows that as the exhaust gases pass backwards they rise rapidly, so that they reach the ceiling about 8 to 16 ft back of the car. By placing kerosene in the intake manifold of the car, dense white smoke was made to accompany the exhaust gases from the car, showing clearly the rise in exhaust gases as indicated by this survey. Fig. 9 is a photograph of this rising smoke.

Survey 26 of Floor 3E by the Pyrotannic Acid Method, April 1, 1932, 3:00 to 3:10 P.M.

Conditions of ventilation and car activity for this survey were identical with those of Survey 25 made with the recorder, with the exception that samples were taken at the ceiling, breathing line, and floor at Stations 1, 3, and 5, for analysis by the pyrotannic acid method. The results are given in Table 4. This survey shows average concentrations at the ceiling, breathing line, and floor of the three stations of 1.3, 1.43, and 0.67.

Survey 27 of Floor 3E by Pyrotannic Acid Method, April 2, 1932, 12:20 to 12:29 A.M.

This survey was made under the same conditions of ventilation and car activity as Survey 28, excepting that two cars were idling as indicated by *X* and *Y*, Fig. 3, and samples were taken for the pyrotannic acid method of analysis. The results are given in Table 4, and show practically the same tendency as those of Surveys 26 and 28 made by the U. S. Bureau of Mines Carbon Monoxide Recorder, excepting that Survey 27, made with two cars idling, shows a slightly lower average concentration than Survey 26.

Survey of First Floor of the Building with Recorder.

An additional survey was made of the carbon monoxide concentration on the first floor of the building to learn whether leakage from the garage resulted in carbon monoxide vitiated air in the building. A trace of carbon monoxide was found which corresponded in magnitude with the concentration found in the city street.

Discussion of Results of Surveys

The surveys in the Grant Building Garage show that with normal car operation the ventilation capacity provided by the fans is ample or excessive, and that the amount of air supplied probably can be reduced. Particularly could such reduction in ventilation be made with safety if some instrument for indicating the concentration of carbon monoxide were available to the operating engineer, such as the Carbon Monoxide Recorder developed by the U. S. Bureau of Mines which was used in this survey. A number of these instruments are in use for control of carbon monoxide concentrations in the Holland Tunnel and elsewhere. The cost of the recorder however, makes doubtful the feasibility of its use, excepting in large establishments.

An indicating instrument based upon the same principles which is less costly has been developed recently. This smaller instrument is limited in sensitivity to

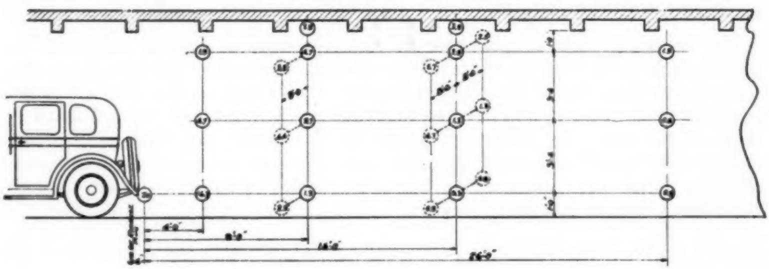


Fig. 8. Survey Behind Idling Car Showing Carbon Monoxide in Parts per 10,000

½ part carbon monoxide in 10,000, which should be sufficiently accurate for the purpose.

Practically all the surveys made either with the recorder or by the pyrotannic acid method of analysis show a higher average concentration of carbon monoxide at the ceiling than at the floor. This is particularly true of surveys with normal ventilation and car operation, or with idling cars. It does not noticeably hold true, however, for conditions where a large number of cars are driven through the roadways of the different floors. A ramp garage differs widely from an ordinary one-room garage space, regardless of size, in that, in the former many cars drive through any given floor, compared with the number of cars idling or traveling at very low speed, while in the latter the reverse is generally true. Hence, the tendency toward high concentrations of carbon monoxide at the ceiling is interfered with to a greater extent by the mixing effect of moving cars in the ramp garage than in the single room garage.

The conclusions reached in the companion report[3] as a result of visual observations of smoke and surveys in the one-floor garage, namely, that exhaust gases from an automobile rise quickly upon leaving a car due to their higher temperature, is borne out by the survey behind a stationary car and the

[3] Carbon Monoxide Distribution in Relation to the Ventilation of a One-Floor Garage, by F. C. Houghten and Paul McDermott (A. S. H. V. E. Transactions, Vol. 38, 1932, p. 425).

photograph in the Grant Building of the exhaust gases rising behind a car. However, these surveys indicate that comparatively little advantage can be gained by upward ventilation on the different floors of a ramp garage, because of the fact that all of the different floor spaces are interconnected, and also because of the fact that the cars driving through the roadways on the different floors tend to mix the air from the ceiling to the floor.

The surveys with a large number of cars operating on the central floors indicate a marked tendency for carbon monoxide laden air to rise to the floors above, with practically no tendency for the reverse or a downward diffusion to floors below those on which the carbon monoxide is being produced. This may be a characteristic of a ramp garage with interconnecting floor spaces, although it is more likely the result of unbalanced air supply and exhaust from the garage.

A survey with smoke indicated a tendency throughout the garage for upward flowing air currents between all floors. The greater quantity of air sup-

Table 5. Vertical Distribution of Supply and Exhaust Air

Floor	Height Center of Outlet Above Floor	Air Volume from Supply Riser No. 7[a] in Cu Ft per Min.	Height Center of Inlet Above Floor	Air Volume to Exhaust Duct No. 3[a] in Cu Ft per Min.
	82	435	60	2414
1	10	427		
	88	370	78	2363
2	10	416		
	88	450	78	3358
3	10	480		
	88	540	78	3642
4	10	548		
	88	568	78	4319
5	10	550		

[a] See Table 2.

plied to the garage over that exhausted, as indicated by the fan deliveries, will account for this upward flow of air. An attempt was made in the design of this garage ventilating system to deliver like quantities of air to all floors. A survey of Supply Riser 7 and Exhaust Duct 3, indicates the extent to which this was accomplished, for these two ducts. These data appear in Table 5.

If air is supplied uniformly to all of the floors and exhausted uniformly from all of the floors, and if the supply is greater than the exhaust, then the excess of supply must necessarily find an exit, which would naturally be upward to the street, giving a vertical air current through the different floor levels of the garage. That this was the case was evidenced by the fact that a high rate of air leakage took place between the garage space and the other part of the basement to the eastward, separated by the unplastered hollow tile partition. Every crack and crevice through this partition showed a high rate of air leakage.

Because of this upward circulation of air resulting from a higher rate of air supply, it is not possible to conclude that air laden with high concentrations

of exhaust gases and carbon monoxide tends to rise to the upper floors because of lower density resulting from higher temperature. However, the fact that the highest concentrations of carbon monoxide in the upper floor spaces where it was not being produced were found near the upper ceiling, rather than uniformly distributed through such floors, would still indicate the possibility of gravity circulation upward of high concentrations of carbon monoxide.

The extent to which change of air from one floor to another takes place is indicated by Surveys 27 and 28, with no ventilation, either supply or exhaust, on the third floor east, but with normal ventilation on other floors of the garage. The concentrations of carbon monoxide found on Floor *3E* for this condition

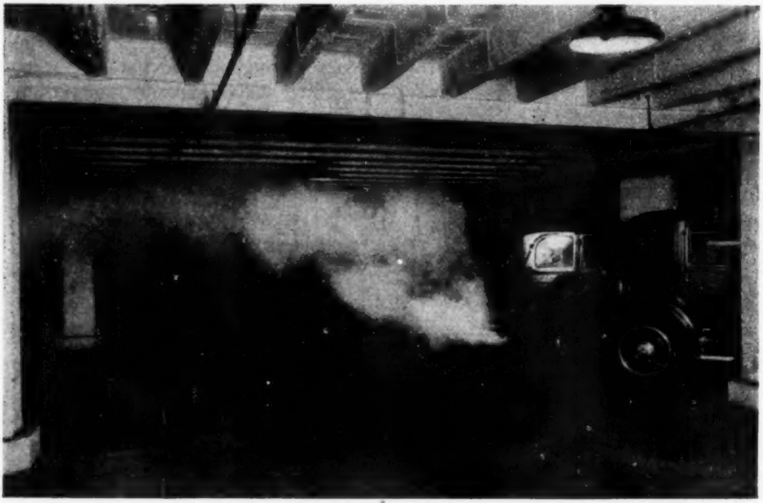


FIG. 9. PHOTOGRAPH OF AUTOMOBILE EXHAUST. (SMOKE PRODUCED BY PUTTING KEROSENE INTO INTAKE MANIFOLD)

were little or no higher than found on this floor with normal ventilation and the same car activity. The reason for this is found in the observed distinct circulation of air into the floor space from below and out at the ceiling.

Surveys 5 and 22, made with no air supply in the southeast corner, indicate the extent to which concentrations of carbon monoxide increased in this corner due to a dead condition or lack of ventilation. While a higher concentration was usually found in this corner when it was not ventilated than was found in the opposite ventilated corner, this difference did not always exist; neither was the difference very great.

The ability of the ventilating system to rapidly bring down the concentration of carbon monoxide, even when it had been increased artificially by running a number of cars and shutting off the ventilating system, is shown in Fig. 7. By taking into account the volume of the garage, the number of cars operating, the amount of air handled, and the rate of increase or decrease in the average

carbon monoxide concentration in the garage at any time, it is possible to compute the rate at which carbon monoxide was being delivered to the garage and the rate per car.

The rate of change in volume of carbon monoxide in the garage with time is given by the equation:

$$\frac{dQ}{dt} = V_{co} - Q\frac{V_v}{V_g} \tag{1}$$

where

$Q =$ volume in cubic feet of carbon monoxide in garage at any time t

$V_g =$ volume in cubic feet of garage

$V_v =$ volume in cubic feet of air supplied per minute

$V_{co} =$ volume in cubic feet of carbon monoxide added to the garage per minute

For the simple case where V_v and V_{co}, or the rate of air change and the rate of carbon monoxide production, are constant the solution of Equation 1 is:

$$Q = \frac{V_g}{V_v}V_{co}\left(1 - e^{-\frac{V_v}{V_g}t}\right) + Q_0 e^{-\frac{V_v}{V_g}t} \tag{2}$$

where

$Q_0 =$ the volume of carbon monoxide in the garage at the beginning of the time considered, or when $t = 0$.

Equilibrium between rate of air change and rate of carbon monoxide addition to the garage will theoretically be reached in infinite time, or when $t = \infty$. Substituting this value for t in Equation 2 gives the following expression for the volume of carbon monoxide in the garage when equilibrium has been established with any rate of carbon monoxide production and air change:

$$Q = \frac{V_g}{V_v}V_{co} \tag{3}$$

If no ventilation or air change is provided the rate of carbon monoxide production is given by the equation:

$$V_{co} = \frac{Q - Q_0}{t} \tag{4}$$

Two sections of the curve, 9:05 to 9:16 P.M. and 10:04 to 10:24 P.M., Fig. 7, giving the carbon monoxide concentration on the evening of March 25, give conditions from which carbon monoxide production might be calculated by Equation 2. The first of these must be discarded because of unsatisfactory operation of the recorder pen at the time. Using the section of the curve between 10:04 and 10:24 P.M. the values of the various terms in Equation 2 become:

$$\begin{aligned} Q &= 140 \\ Q_0 &= 357 \\ V_g &= 700{,}000 \\ V_v &= 60{,}000 \\ e &= 2.71828 \\ t &= 10{:}24 - 10{:}04 = 20 \end{aligned}$$

Solving Equation 2 for V_{co} gives 7.81 cfm of carbon monoxide added to the garage, and 468.6 cfh for 17 cars, or 27.6 cfh per car.

Two sections, 8:10 to 8:50 and 9:40 to 10:00, of the curve, Fig. 7, may be used for calculating the rate of carbon monoxide production from Equation 4, or for a condition of no ventilation while the carbon monoxide concentration is building up from running cars. For these periods the values of the terms in Equation 4 become:

	8:10 to 8:50 P.M.	*9:40 to 10:00* P.M.
$Q =$	507.5	358.4
$Q_o =$	43.4	183.4
$t =$	40.	20.

Solving Equation 4 for V_{co} gives 11.6 cfm of carbon monoxide added to the garage, giving 696.0 cfh for twenty cars or 34.8 cfh per car between 8:10 and 8:50, and 8.75 cfm, giving 525.0 cfh for 17 cars or 30.9 cfh per car between 9:40 and 10:00.

The values of 27.6, 34.8, and 30.9 cfh for the rate of carbon monoxide production per car compare favorably with 29.4 and 31.7 found in the single-room garage previously reported. They are lower than the values found by the U. S. Bureau of Mines about ten years ago. This discrepancy can be accounted for by change in motor design or quality of gasoline during the past ten years, or it may be due to air change in the garage other than that measured.

ACKNOWLEDGMENT

The authors acknowledge the kind cooperation of C. E. Traubert of the U. S. Bureau of Mines and H. B. Meller of the Mellon Institute of Industrial Research who aided in making available, calibrating and operating the U. S. Bureau of Mines Carbon Monoxide Recorder, and to W. J. Strassburger and Verne W. Hunter of the Grant Building, who made this building available for the study and assisted in many instances to make the study a success.

DISCUSSION

W. C. RANDALL (WRITTEN): It is interesting to note that the surveys confirm some of the findings in Ramp Garages as reported in Airation Studies of Garages, A.S.H. & V.E. TRANSACTIONS, Vol. 35, 1929, in that the carbon monoxide is, in general, very evenly distributed and of low content, except possibly at the rush periods morning and night. That would indicate the delivery of air in comparable quantities by natural ventilation in our studies and by mechanical ventilation in the paper under discussion. They both indicate lack of pockets for the reasons similarly reported in the two papers, that is chiefly the movement of the cars and the merging of the floors by ramps.

I would suggest the extension of the formulae and mathematical computations to allow a garage manager to predetermine the mechanical ventilation required, based on the car hours running time for peak and non-peak loads. The low values of carbon monoxide per running hour explain why some of the low concentrations were found compared with our expectancy, using 1½ cu ft per car minute. I think the values found would not be adequate for winter conditions when carburetors are normally adjusted for a richer mixture.

No. 936

INVESTIGATION OF AIR OUTLETS IN CLASS ROOM VENTILATION

By G. L. Larson [1] (*MEMBER*), D. W. Nelson [2] (*MEMBER*) and R. W. Kubasta [3] (*NON-MEMBER*), Madison, Wis.

This paper is the result of research conducted at the University of Wisconsin in cooperation with the A. S. H. V. E. Research Laboratory

THE purpose of this investigation was to study the influence of outlet vents on class room ventilation. Air infiltration tests have shown that most wall constructions offer paths for air travel. This is especially true around windows and doors. Plastered walls have been shown to be very resistant to air passage, but along the edges of window and door openings and at baseboards, there are chances for the passage of considerable quantities of air. Infiltration depends upon a difference in pressure; therefore, planned outlet vents have an influence on the infiltration into the room. If smaller outlet vents were to be used or if outlet vents were to be eliminated entirely, the consequent building up of pressure might be sufficient to overcome all infiltration into the heated spaces.

Description of Buildings and Ventilating Systems

The tests were conducted in three school buildings: Dudgeon School of the Madison system, the Mechanical Engineering Building and the main Engineering Building of the University of Wisconsin. Dudgeon School is one of the new buildings of the Madison city system and is equipped with unit ventilators. The units are operated on direct current thus rendering variation in capacity very convenient. The class and drawing rooms of the third floor of the Mechanical Engineering Building are ventilated by a central fan system. A typical class room was selected for tests. In addition to testing with a central fan supply, a unit motor-blower and also a unit ventilator were set up in this room to determine the influence of the type of supply on the results obtained in the outlet vent study. A long series of observations was carried on at the main Engineering Building to determine the variations in air supply with open windows under various temperatures and wind movements. These observations were made with and without outlet vents in the two rooms selected.

[1] Professor of Steam and Gas Engineering and Chairman, Dept. of Mechanical Engineering, University of Wisconsin, Madison, Wis.
[2] Assistant Professor of Steam and Gas Engineering, University of Wisconsin, Madison, Wis.
[3] Instructor in Steam and Gas Engineering, University of Wisconsin, Madison, Wis.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Milwaukee, Wis., June, 1932, by D. W. Nelson.

FIG. 1. DUDGEON SCHOOL TEST ROOM

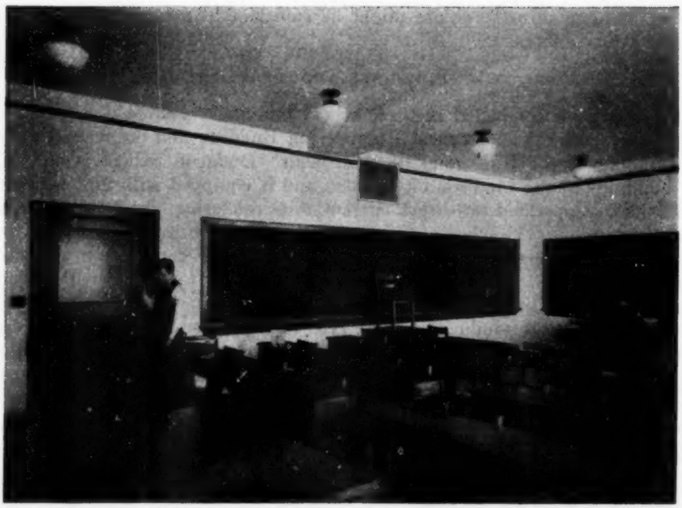


FIG. 2. TEST ROOM AT THE MECHANICAL ENGINEERING BUILDING

The planned outlet vents at Dudgeon consist of an opening in each class room at the floor line from which a sheet metal duct extends vertically upward about 8 ft and discharges into the corridor through a wire grille. Venting from this point takes place through grilles in toilet room doors and also through lockers located in the corridors. The building is a single story in height. The size of the vertical duct in the class room and the grille is 12 in. by 21 in. The floor dimensions of the class room are 22 by 29 ft. Fig. 1 shows a general view of this school room.

The venting of class rooms and drawing rooms in the Mechanical Engineering Building is through louvers placed in the doors leading into the corridors.

Fig. 3. Open Window Test Room at the Engineering Building

Venting from the corridors is largely through roof ventilators. The dimensions of the typical class room selected for observations are 23 by 32 ft. The gross area of the outlet vent is 5.4 sq ft. Fig. 2 shows this outlet vent.

Two rooms were selected in the main Engineering Building for the open window tests, one on the south side and one opposite this one on the north side of the building. The floor area of the south room was 700 sq ft, and the gross outlet grille area was 5.4 sq ft. The floor area of the north room was 624 sq ft, and the gross outlet grille area was 5.4 sq ft. The vent ducts from various class rooms unite and discharge above the roof. Fig. 3 shows the south room.

Calibration Testing

Anemometers were used for air measurements in all cases except with the air delivery from the motor-blower. It was considered advisable to calibrate the

particular anemometer used on each grille or louver or open window both for correct registration and for the type of opening on which it was to be used. Two 3-in., two 4-in., and a 6-in. anemometer were used during the program.

In order to determine the true air flow as compared to the anemometer registration, the anemometers were moved through a 26-ft diameter circle on the end of a whirling arm. Electrical contacts and solenoids mounted on the whirling arm were used to start and stop the anemometer registration. The corrections determined for anemometers that had been used for several years checked closely with the manufacturer's determinations. The anemometers were cleaned and oiled before testing.

In all cases this anemometer correction was applied to observed readings. In addition it was considered necessary to determine the correction to be applied to the readings for the type of opening and air flow with which the anemometer

FIG. 4. CALIBRATION OF ANEMOMETER ON UNIT VENTILATOR

was used. This was done for the unit ventilators installed at Dudgeon and the unit ventilator that was temporarily set up in the test class room at the Mechanical Engineering Building. Fig. 4 shows the calibrating arrangement for the latter. A duplicate of the outlet duct in the class room at Dudgeon was set up as shown in Fig. 5, and the calibration of the anemometer was determined for use on this outlet grille. Fig. 6 shows a door of a class room in the Mechanical Engineering Building in position for calibration of the 6-in. anemometer on the outlet louvers.

The anemometer was held against the face of the grille or louver in every case. Air was supplied by a motor-blower and the air measurement was made by Pitot tube traverses. The unit ventilator fan was operated during calibration runs. The air supply was varied at the motor-blower inlet and the corresponding static pressure at the unit ventilator inlet was observed. These static pressures and the calibration factors for the two unit ventilators tested are shown in Fig. 7. The unit ventilator at the Dudgeon class room was rated at 1,350 cfm. The calibration factor determined at zero inlet static pressure was

0.73. The correct air flow under this normal inlet pressure was found to be 1,080 cfm without the filter and 950 cfm with a clean filter. The unit ventilator installed for the tests in the class room in the Mechanical Engineering Building was rated at 1,200 cfm. At zero inlet static pressure, the calibration factor was 0.87. The correct air flow was determined as 744 cfm without the filter and 615 cfm with a clean filter. The Dudgeon unit had a higher velocity of discharge than did this unit. The lower calibration factor of the Dudgeon unit

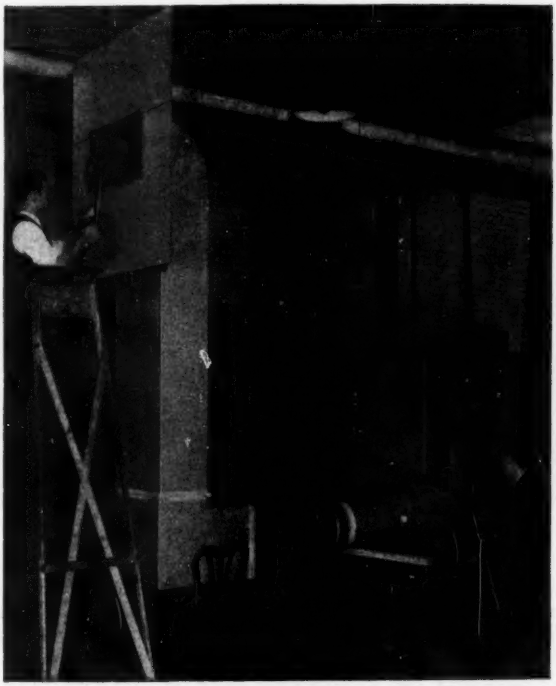


Fig. 5. Calibration of Anemometer on Outlet Duct Grille

is accounted for by the fact that the lower row of grille openings was practically inactive. It seems characteristic of anemometers that they measure higher than the average velocity when placed in a position of widely varying velocities. All calibrating was done with the grilles marked off in even squares and the anemometer was kept in contact with the grille the same length of time at each square. The average of the gross and net area was used in each case in figuring air volumes.

The results of the calibration tests for the grille on the outlet duct at Dudgeon and for the louver opening in the door at the Mechanical Engineering Building

are shown in Fig. 8. When the 3-in. anemometer was calibrated on the outlet duct, the grille was divided up into four horizontal rows. It was found that the velocity was zero in the two bottom rows. When only the upper half was considered and the average of the net and gross area of this was used, the calibration factor was 1.00. This is in agreement with the results previously determined by the Society.[4]

In calibrating the 6-in. anemometer on the louver opening, the anemometer was held against the louvers and parallel to the face of the door. The louvers are ⅛ in. thick and are placed at an angle of 45 deg. The gross area was based on the width and height through which air passed. In this way the space between the uppermost louver and the edge of the opening margin was not included since it was closed to passage of air at the inner face of the door. The

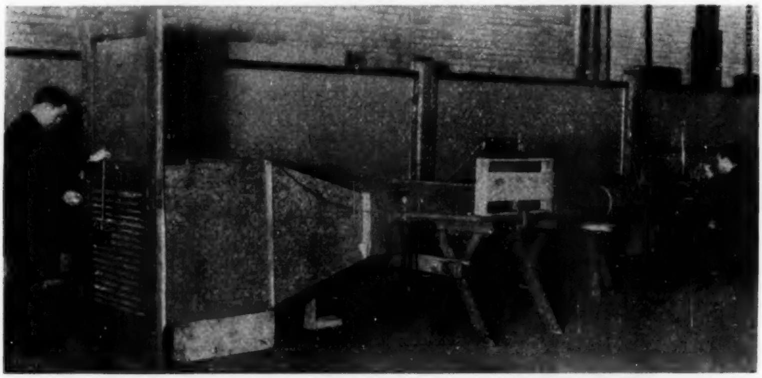


FIG. 6. CALIBRATION OF ANEMOMETER ON LOUVER DOOR

net area was determined by subtracting the area of the vertical dividing bar and the area of the louvers as determined by their length and the thickness perpendicular to their own surfaces. The average of the gross and net area was used in determining the calibration coefficient. The approach section in the duct for calibration had an area about twice the average area of the louver opening. The factor for most of the range varied from 0.80 to 0.85.

Based on the results obtained on the wire outlet grille pictured in Fig. 5 and the previous calibrating work done by the Society[4] on such grilles, it was decided to be unnecessary to calibrate the wire inlet grille of the class room tested in the Mechanical Engineering Building. A calibration factor of 1.00 was accordingly used.

RESULTS AT DUDGEON SCHOOL

A preliminary set of readings in the class room at Dudgeon School proved that sealing of all openings did not reduce the capacity of the unit ventilator

[4] See Measurement of Flow of Air Through Registers and Grilles, by L. E. Davies, A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930 and Vol. 37, 1931.

materially. It was decided to seal the cracks in the mullion spaces before taking further readings. An attempt was also made to seal plaster cracks in the ceiling that would allow air to enter the attic space and to seal cracks in a side wall that allowed air passage to an adjoining toilet room.

One of the series of tests made after this preliminary sealing of unusual exits for air is shown in Fig. 9. The speed and the unit ventilator output were varied over the range provided by the allowable voltage variation at the motor-generator set. This range was from 87 volts to 137 volts. Normal voltage was 125 volts. Two sets of tests were made, one with the outlet duct open and another with it completely sealed. The results for each determine practically a straight line variation between voltage and air delivery. At the lowest air delivery the influence of the outlet duct is shown to be 63 cfm. This increased to about 72 cfm at the maximum output. In relation to the capacity with the

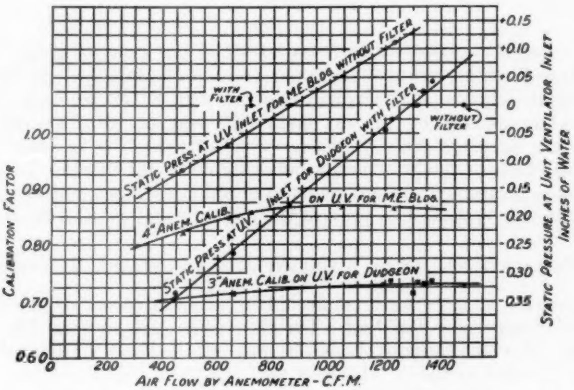


Fig. 7. Calibration Factors for Anemometers on Unit Ventilator Grilles

outlet vent open the capacity with the outlet vent sealed was about 90 to 92 per cent. The capacity at normal voltage was 792 cfm without the outlet vent. The values obtained in some other tests at normal voltage of this same unit were 820 cfm and 755 cfm, respectively. The filter which was thoroughly cleaned was kept in place during all observations. The calibration tests on a similar unit in the laboratory showed 14 per cent greater capacity with the filters removed than with them in place. The unit used for calibration in the laboratory was operated by alternating current, otherwise the two units were identical. The capacity obtained with the laboratory unit at zero static pressure with the filter installed was 950 cfm as compared to 820 cfm for the unit tested at Dudgeon School.

In a series of tests at Dudgeon when 820 cfm was supplied by the unit ventilator, 55 per cent left the room by way of the planned outlet vent. When this outlet vent was sealed, the air supplied by the unit ventilator was reduced to 758 cfm or 92 per cent of normal capacity. Sealing the sash perimeters of the

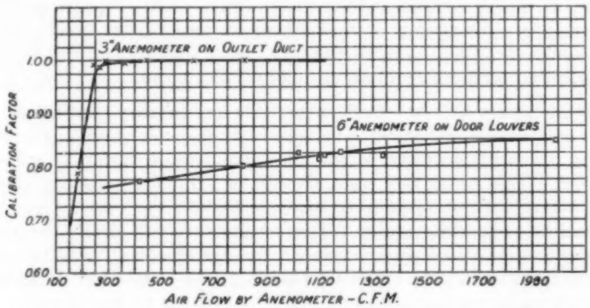


FIG. 8. CALIBRATION FACTORS FOR ANEMOMETERS ON OUTLET VENTS

windows reduced the capacity to 700 cfm or 85 per cent of normal capacity. The additional sealing of the door and transom perimeters reduced the air volume supplied by the unit ventilator to 590 cfm or 72 per cent of normal capacity. This amount of air was finding its way out of the room through various small openings such as plaster cracks, under the baseboard, around steam pipes, through the floor and around the door and window frames. The recirculating openings, steam and return connection openings and all joints were sealed in the unit ventilator to prevent the measurement of recirculated air.

OBSERVATIONS AT THE MECHANICAL ENGINEERING BUILDING

Similar series of tests were conducted in a typical class room of the new Mechanical Engineering Building. The class rooms are ventilated by a central fan system. A unit ventilator, however, was installed and the central fan inlet sealed for a series of tests. This unit was equipped with a rheostat so that a speed variation was possible. Two sets of tests were made at different motor

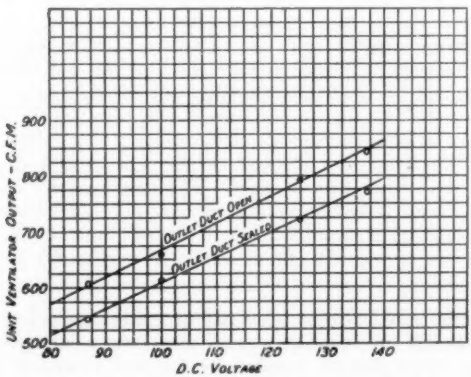


FIG. 9. DUDGEON UNIT VENTILATOR OUTPUT

speeds, one with the central fan running, so that air was being supplied to adjoining class rooms and the corridor as under normal operation, and another series with the central fan shut off. The difference in the two shows the effect on the unit ventilator of the pressure exerted in the corridor and adjoining class rooms by the central fan. The outlet vent area is large and leads directly from the corridors to the roof so that the pressure in the corridor was about one-half that in Dudgeon School.

The results of tests of the class room when equipped with a unit ventilator are shown in Fig. 10. The abscissa used is the area involved in escape of air from the room. The distance *A* to *B* represents the 5.13 sq ft of area of the louver opening in the door; the distance *B* to *C* represents the crack under and around the door perimeter; and the distance *C* to *D* represents the approximate area of the window sash perimeter crack.

The points at *A* with none of the outlet openings sealed show that at every

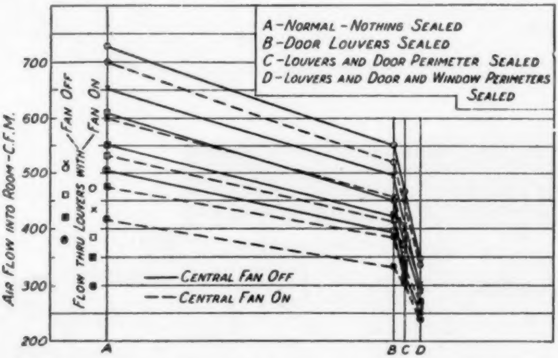


FIG. 10. UNIT VENTILATOR OUTPUT AT MECHANICAL ENGINEERING BUILDING UNDER DIFFERENT OUTLET CONDITIONS

motor speed there was less air delivery with the central fan on than with the fan off. In every case but one, the difference in the air passing through the louvers accounted largely for the reduced air delivery. This difference as measured at the unit ventilator grille ranged from 28 cfm at the highest delivery to 88 cfm at the lowest motor speed and air delivery. In per cent of the deliveries with the central fan off these would be 4 per cent and 17.5 per cent. The average reduction due to the effect of the central fan was 64 cfm. With the louvers sealed the average effect was reduced to 41 cfm. With further sealing, such as sealing the ½-in. crack beneath the door, the difference became very small as shown at *C* and *D*.

The maximum air delivery secured with the unit ventilator installation averaged 728 cfm. On the calibration runs in the laboratory on this same unit ventilator, 744 cfm was secured at a zero static inlet pressure. This decrease in capacity was probably due to the building up of pressure in the room and to the resistance offered by the special inlet duct required so as to bring the air in through the bottom row of window panes.

Upon sealing the louvers the air delivery of the unit ventilator was reduced from 728 cfm to 552 cfm or a 24 per cent reduction. Additional sealing of the door perimeter reduced the air delivery to 466 cfm, which is 64 per cent of the normal delivery. Further sealing of the window sash perimeters reduced the air delivered by the unit ventilator to 346 cfm, a reduction to 47.5 per cent of the normal delivery.

The slope of the lines from *B* to *C* and *C* to *D* seems to indicate that they would meet if extended downward approximately when the air flow into the room would be zero at 0.9 sq ft to the right of line *D*. This would indicate very roughly that the various minute openings for escape of air from the room had a total area less than one square foot. An inspection of the room makes

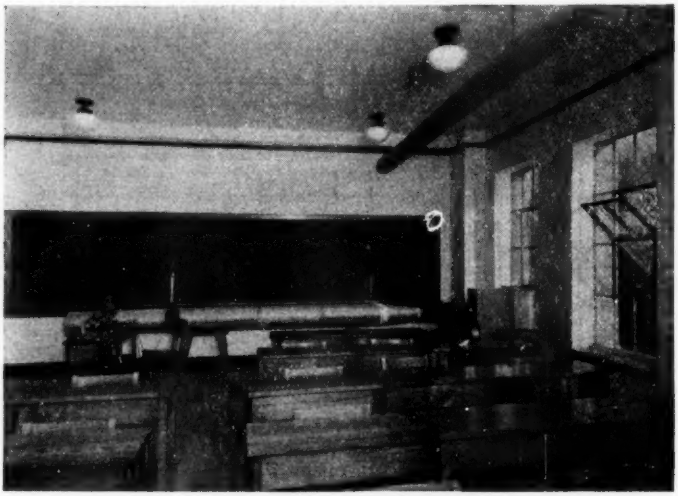


FIG. 11. MOTOR-BLOWER ARRANGEMENT IN MECHANICAL ENGINEERING BUILDING CLASS ROOM

it seem improbable that the openings remaining after sealing the louvers, the door perimeter and the window sash perimeters, would total more than the perimeter of the windows. No plaster cracks were noticeable. The windows are set in such a way that frame leakage would seem to be negligibly small. The plastering extends to the concrete sub-floor quite uniformly. The steam pipe openings where an overhead main goes through two tile partition walls, where the steam supply branch comes through a tile partition and where the return connection passes through the floor to the laboratory space below would seem to be the main avenues of exit. During the runs while the central fan was in operation, the pressure in adjoining class rooms would oppose at least partially the passage of air around the steam pipes into these rooms. However, it would still be possible for air to enter the hollow tile in these partitions and reach the attic space. It appears unlikely that the area of these minute openings would total over 0.3 sq ft which is the area estimated for the window

crack perimeter. Figs. 3 and 11 show some of the construction details of the room.

Tests with Motor-Blower Supply

The unit ventilator capacity is sensitive to the building up of pressure. The calibration tests in the laboratory showed this by a decrease in capacity of about 150 cfm for an increase of 0.05 in. water in pressure drop across the unit. In order to study the effect of outlet vents under different conditions of air supply, the motor-blower used for calibrating of anemometers on grilles and louvers was moved up into the room and arranged for an outside air supply. The air delivery was measured by Pitot tube traverses. Fig. 11 shows this temporary arrangement at the rear of the class room. No provision was made for heating the incoming air as was done in the case of the unit ventilator installation.

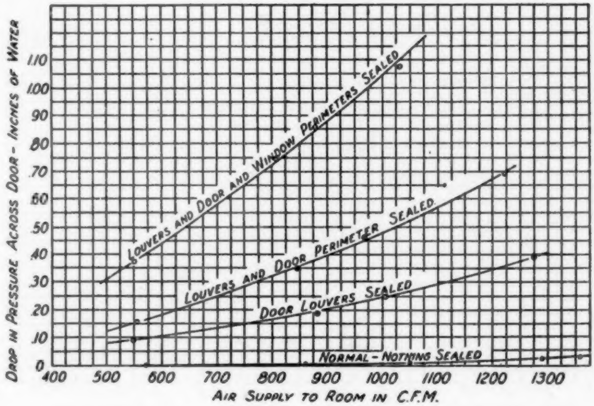


Fig. 12. Relation of Pressure Drop and Air Supply from Motor-Blower for Different Outlet Conditions

In both cases the air volumes were corrected to 70 F before making comparisons.

In a typical test of the motor-blower, the air delivery to the room was 910 cfm with the outlet vent open. The amount of this measured at the door louvers was 73 per cent. Sealing of the door louvers resulted in only a three per cent reduction. Sealing of the door perimeter reduced the capacity to 89 per cent. Tests with the blower inlet partially throttled so as to obtain lower air delivery showed even less falling off in air delivery with progressive sealings. In the lowest run taken, 545 cfm were obtained with nothing sealed; with the outlet vent, door and window perimeters sealed, the capacity was reduced only to 520 cfm. Fig. 12 shows the drop in pressure from the class room to the corridor as plotted with the capacities of the various tests. This pressure drop was less than 0.01 in. water with the delivery of 1000 cfm. With the sealing of the door louvers, the same air delivery resulted in a 0.25-in. drop in pressure. Sealing of the door perimeter in addition resulted in a pressure drop of 0.48 in.

for the delivery of 1000 cfm. The same air delivery with the additional sealing of the windows made a pressure drop of 1.04 in. water across the wall between the room and the corridor. The outlets from the corridor to the roof are large so that these pressure drops approach the pressure against which the fan was working. Measurements of the electrical input on this motor-blower showed little difference in current consumption for a delivery of 1000 cfm at 0.02 in. water and at 1.04 in. water. This fan is a pressure fan capable of working against 6 in. of water pressure and consequently was working at a low efficiency at 1.04 in. and still lower at 0.02 in. The power consumptions with this pressure blower do not give the correct power cost comparisons for a system designed for delivering air into tight rooms without outlet vents, compared to that found with the ordinary vented system.

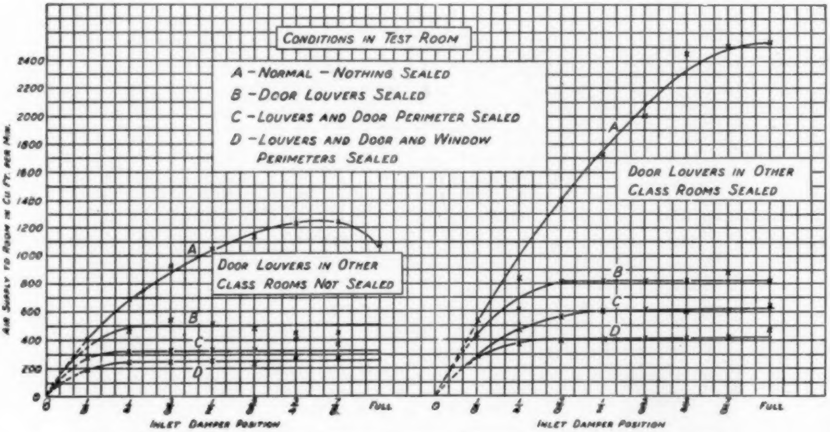


FIG. 13. CENTRAL FAN SUPPLY UNDER DIFFERENT OUTLET CONDITIONS

CENTRAL FAN SYSTEM

The tests described so far were made with a unit ventilator and with a motor-blower with the inlet of the central fan system sealed. The central fan was used in most of these runs to provide normal conditions in the corridor and in the two adjoining class rooms. Because the unit ventilator was sensitive to the building up of pressure, in one series the central fan was not run so as to reduce the pressure in the corridor against which the unit must discharge.

Further observations were made using the central fan for supplying the air to the class room under test. Two series of tests were made, one with the outlet louvers open in all other ventilated rooms and the other with these other louvers sealed during the period of test. Fig. 13 shows the results of these tests. The maximum capacity delivered to the room with the louvers in the other class rooms open was 1250 cfm. When these other louvers were sealed the maximum delivery to the test class room with the outlet louvers of

this room open was 2530 cfm. This shows the diverting of air away from the rooms where the resistance has been increased.

In the case of the outlet vents of other class rooms not being sealed, the sealing of the door louvers in the test class room reduced the capacity from 1250 cfm to 500 cfm, which is a reduction to 40 per cent. The further sealing of the door reduced the air delivery to 26 per cent and the additional sealing of the windows reduced it to 20 per cent. These reductions are more than for the unit ventilator or motor-blower. This shows again the diverting of the air to other class rooms as resistance to out-flow is increased in the test room.

The sealing of the door louvers in the test room in the series where the door louvers of all other class rooms were sealed reduced the maximum air volume from 2530 cfm to 820 cfm, a reduction to 32 per cent. The additional sealing of the door perimeter reduced the capacity to 615 cfm, which is 24 per

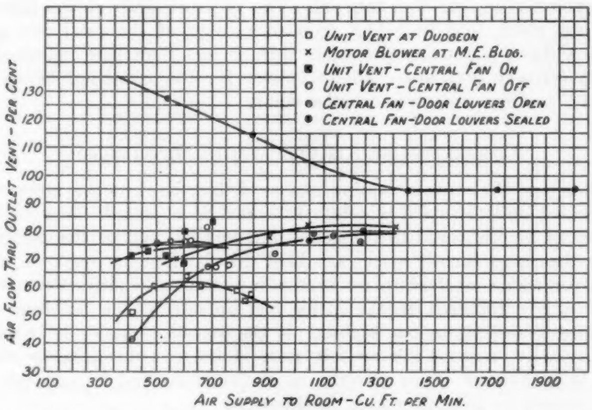


FIG. 14. RELATION OF OUTFLOW THROUGH PLANNED OUTLET VENTS TO AIR SUPPLY

cent of the delivery with nothing sealed. The further sealing of the windows reduced the delivery to 16 per cent.

Readings were taken at carefully set damper positions from full open to closed by eighths. Closing of the dampers on the two runs with nothing sealed in the test room resulted in a decrease in air volume; on the various sealed runs the damper had little effect until nearly one-fourth closed.

It should be emphasized that these figures do not give the correct comparison between two ventilating systems installed in the same rooms, the two being identical except for the exits of air. In the cases cited, as the resistance was built up in the test room by the sealing of exits, the air was diverted to other class rooms. For a true comparison, the same sealing should be carried out in every ventilated room at the same time. However, the correct comparison is available for these two systems, one with and one without planned outlet vents. In Fig. 13 the maximum air delivery was 1250 cfm with the louvers

open in all class rooms. This is shown in Curve *A* on the left half of the figure. In Curve *B* on the right half of the figure is shown the air delivery with the same inlet damper setting of ⅞. This delivery was 820 cfm which is 66 per cent of that obtained when outlet vents in the form of door louvers are provided. Had a lower damper setting been chosen for this comparison the per cent would have been larger as the damper reduced the unsealed delivery considerably while not affecting the sealed delivery over a wide range.

A measurement of the air delivery from all central fan inlets with all louver doors not sealed totaled 17,514 cfm. With the inlet damper wide open the test class room received 1250 cfm. This is about 7 per cent of the total air for the system. With all louvers, the outlet vents from the ventilated rooms sealed, the total capacity of the system as measured at the inlets to the various rooms was 15,130 cfm. This is 86 per cent of the delivery obtained with all door louvers open. The discharge pressure at the fan increased from 0.21 in. to 0.28 in. upon sealing the door louvers. The pressure drop on the inlet side of the fan was about 0.38 in. The ratio of change of the total fan pressures with and without planned outlet vents then is less than that indicated by the change in delivery pressure. This accounts for the reduction being only 14 per cent for sealing of the room outlet vents.

COMPARISON OF AIR QUANTITIES DISCHARGED

The ability of the various equipments to deliver against the increased pressure of sealed outlets varied widely. It would be expected, however, that the proportion of air delivered to the room that left by way of the outlet louvers would be the same for all systems. That this is not true must be attributed to the inaccuracies of testing or to the influence of infiltration. With the pressures nearly balanced on the inside and outside of the building wall, the sash perimeters form one exit for air. If the pressure outside is in excess due to wind velocity, exit of air by this path is reduced, brought to zero or is replaced by a delivery of air into the room that must find its way out through other exits such as the louver door in this case.

Fig. 14 shows the air flow through the outlet vent plotted in per cent of the air supplied to the room by the different units used. Included on this sheet are also the results of the tests of the Dudgeon School class room. The outlet vent in this room is more restricted than that in the Mechanical Engineering Building class room, so it is to be expected that the curve shows a lower per cent of the air supply leaving by the planned outlet vent. Further, the small miscellaneous openings are larger and more numerous in this room.

The results shown for the central fan with the louvers open in all class rooms check fairly closely with the motor-blower results. At 1000 cfm, the air flow through the outlet vent was about 80 per cent of the air supply.

In trying to determine more accurately, the two unit ventilator curves, one with the central fan running to give normal conditions in the corridors and one with this fan not running, several check tests were made but the points scattered considerably so that there was a variation of 10 to 15 per cent at equal air supplies. The drop in pressure from the room to the corridor in these runs was less than 0.01 in. water, so that infiltration would not be checked to any extent. A wind velocity of 15 mph exerts a pressure of 0.108 in. water.

With a larger air supply as with the motor-blower and central fan, the pressure builds up, and variable infiltration should affect the points less. At the maximum delivery of the motor-blower the pressure drop from room to corridor was only 0.022 in. water. This indicates that the outlet vent was too large to allow building up enough pressure to prevent infiltration at ordinary wind velocities. With the outlet vents, the louvers in this case sealed, a delivery of 640 cfm as shown on Fig. 12 would build up a pressure sufficient to prevent infiltration with a 15-mph wind velocity. On judging the resistance to infiltration the pressure drop from room to corridor was used. The pressure drops across the outside building wall varied considerably under the influence of the wind. Since the drop in pressure from the corridor to the outside was very low because of the large outlet vents the figures used are considered satisfactory for this purpose.

The curve showing the central fan results with the louver openings in all other class rooms sealed is far removed from the other curves. It would be expected that it would lie a short distance above the other central fan curve because of the lower pressure in the corridor. That it does not is due to the influence of infiltration.

At the lowest air deliveries to the room from the fan inlet 27 per cent more air left the room by the louver opening than the fan supplied. This is 146 cfm more than the 540 cfm supplied by the ventilating system. As the air supply to the room from the fan increased the amount passing through the louvers decreased until 95 per cent of the air supply left by the louver opening. It is probable that the other 5 per cent, amounting to from 70 to 100 cfm, passed through the ½-in. space under the door. The other curves indicate that, if infiltration had not entered to a large extent into the total air entering the room, the air through the outlet vent would be about 85 per cent. The 10 per cent difference between this and the 95 per cent measured at above 1400 cfm would seem to be infiltration and would be from 140 to 200 cfm.

Two additional series of tests were made to check the measurements shown for the central fan on Fig. 12. The tests of the central fan supply with door louvers sealed in all other class rooms as shown on this figure were obtained in the class room with an eastern exposure with a southeast wind having a velocity of 15 mph and a temperature of 46 F. The check tests were made with a 15 mph west wind and an outside temperature of 50 F. The points determined a curve somewhat lower than shown on Fig. 12 for the central fan with louvers open. The results of the check test with the louvers open made the same day were comparable to the results of the tests with the louvers closed in the other class rooms. The amount passing through the outlet vents seems to be considerably influenced by air infiltration.

SUMMARY OF OUTLET VENT STUDY WITH MECHANICAL VENTILATION

Fig. 15 shows a graphical summary of the influence of sealing various exits on the air supplied by the various systems. The maximum deliveries of the unit ventilators have been used, and air volumes in the unsealed runs chosen for the central fan and motor-blower between 900 cfm and 1000 cfm. The differences in the heights of the normal or *nothing sealed* volumes have no particular significance. The motor-blower and central fan inlet were both

capable of supplying considerably more than the quantities shown, but values were chosen close to the range of the unit ventilators tested.

The absence of outlet vents in the Dudgeon School room reduced the capacity only to 92 per cent, whereas this resulted in a reduction to 75 to 77 per cent with the unit ventilator in the Mechanical Engineering Building class room. The central fan, whether it was on or off, exerted little influence on the air delivered by the unit ventilator. This was due to the relatively free discharge of air from the corridor through roof ventilators and the consequent low pressure built up there. Measurements of the pressure drop from the corridor to the outside with an outside temperature of 60 F were 0.009 in. water with the fan off and 0.015 in. water with the fan running.

The greater decrease resulting from the sealing of the outlet vents in the

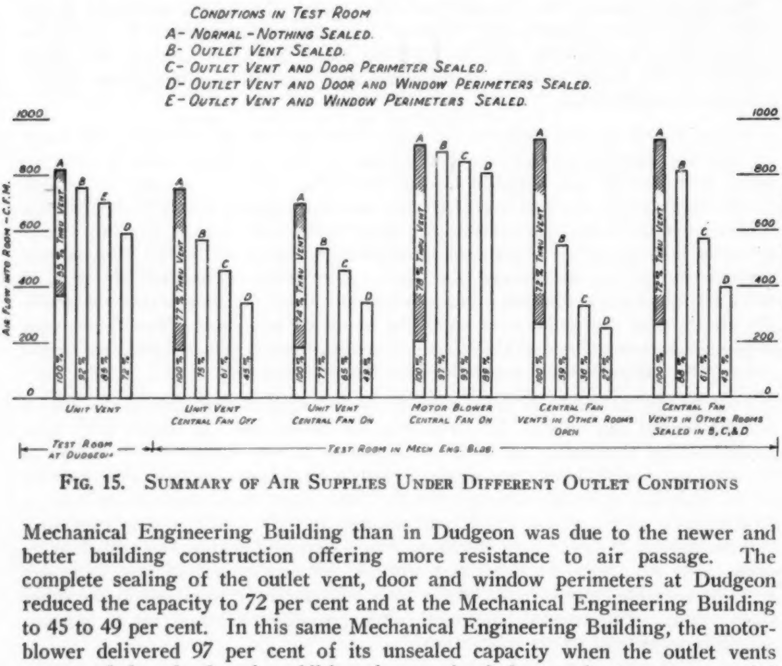


FIG. 15. SUMMARY OF AIR SUPPLIES UNDER DIFFERENT OUTLET CONDITIONS

Mechanical Engineering Building than in Dudgeon was due to the newer and better building construction offering more resistance to air passage. The complete sealing of the outlet vent, door and window perimeters at Dudgeon reduced the capacity to 72 per cent and at the Mechanical Engineering Building to 45 to 49 per cent. In this same Mechanical Engineering Building, the motor-blower delivered 97 per cent of its unsealed capacity when the outlet vents were sealed and when in addition door and window perimeters were sealed the delivery was still 89 per cent.

With the outlet vents in all other class rooms open as was the case of the previously-mentioned comparison from Fig. 15, the air delivery with the central fan was reduced to 59 per cent upon sealing the outlet vents in the test room. This large reduction is due to the diverting of air to other rooms with the increase of resistance in the test room. With the opening of windows in a particular room air is diverted from other rooms to the one with these addi-

tional outlets. When this is done in many rooms at once, the fan discharge pressure may decrease enough to overload the fan motor.

A truer comparison of a central fan system designed with and without planned outlet vents is given in that part of Fig. 15 at the extreme right. The correct comparison should be that of the delivery when the outlet vents are open in the test room as well as in all other ventilated rooms with the delivery obtained when the outlet vents in the test room and all other ventilated rooms are sealed. This prevents the diverting of air because the resistance is built up in all rooms at the same time. Under these conditions, sealing of the outlet vents reduced the air delivery to the test room to 88 per cent, and reduced the total delivery to all of the rooms to 86 per cent. This reduction is less than that with the unit ventilators and more than that with the motor-blower. In the further sealing of window and door perimeters, the same procedure should have been carried on in all other class rooms as in the test room. This seemed impractical to do, consequently the 61 per cent for additional sealing of doors and the 43 per cent for the further sealing of windows are a little lower than they would have been with the more exact procedure. With that procedure, it is to be expected that these percentages would be between those for the unit ventilator and the motor-blower tests, whereas with the procedure used, they are very near to the unit ventilator figures.

The slope of the heights of the air volumes resulting from increased sealings of air exits gives an indication of the tightness of a room. The steeper the slope, the tighter the room.

The air passing out of the room through the outlet vent was 55 per cent of the supply at Dudgeon School. In the various tests at the Mechanical Engineering Building, it varied from 72 to 78 per cent. The larger quantity leaving in the latter building is accounted for in the area of the vent being greater, by its being a door louver opening instead of a vent duct with two 90 deg bends, and the tighter construction of the rooms. As stated in connection with Fig. 14, wind velocity and air infiltration have considerable influence on the amount of air passing through the outlet vents.

Air Distribution in the Class Room

Observations were made on the air distribution with and without planned outlet vents for both the unit ventilator and the central fan. Tests were first made with hydrogen sulphide introduced into the air supply and filter papers dipped in a lead acetate solution placed on 16-in. stands on each of the student tables. The filter paper turned black upon exposure to hydrogen sulphide. The tests were repeated with sulphur bombs which gave off a dense yellow smoke. The smoke method proved to be more satisfactory. The time to reach a given station was longer for the hydrogen sulphide method, and it was difficult to judge the first shade of darkening. Fig. 16 shows the results of the two sulphur bomb tests with the unit ventilator, and Fig. 17 the results for this method with the central fan. The time in minutes to reach each station 4 ft above the floor for the sealed louver runs is shown circled and is shown plain for the unsealed louver runs.

The unit ventilator volumes were 700 cfm in the unsealed and 540 cfm in the sealed run. The corresponding deliveries for the central fan runs were 1080 cfm and 500 cfm.

The distribution seemed very good in every test. There seemed to be no shorting to the outlet vent when unsealed and no undue sluggishness in reaching all locations when the outlet vents were sealed. In every case most of the air travelled across the top of the room from the inlet before dropping

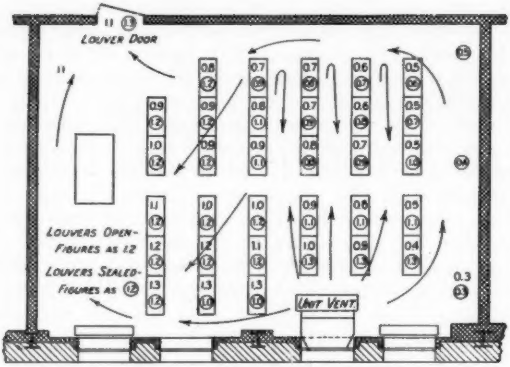


FIG. 16. AIR DISTRIBUTION TESTS WITH UNIT VENTILATOR

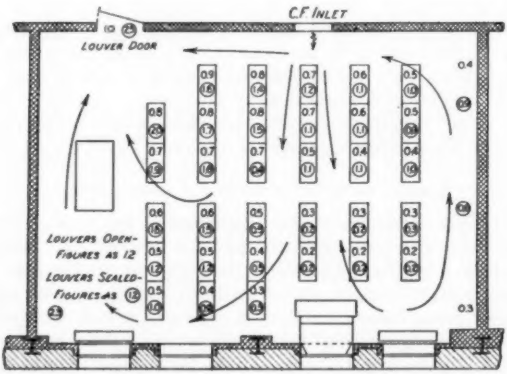


FIG. 17. AIR DISTRIBUTION TESTS WITH CENTRAL FAN

to the 4-ft level. The shortest time recorded to a table station with the outlet vent open and with the unit ventilator supply was 0.4 min, and the longest was 1.3 min. With the outlet vent sealed the shortest time was 0.6 min, and the longest 1.3 min. With the central fan system and the outlet vent open, these times were 0.2 min and 0.9 min. With this system and the outlet vent sealed, these times were 0.2 min and 2.0 min. The time to reach the louver

opening in the door in the unsealed tests was practically one minute in each case. In the sealed test with the unit ventilator, the time to reach the door was 1.3 min. and was 2.5 min with the central fan system. The ½-in. crack under the door offered the definite exit for air at this location when the louver opening was sealed. It is considered that the distribution was very good in each case and that no evidence is shown in Figs. 16 and 17 for or against planned outlet vents.

Open Window Ventilation

Observations have been taken over a period of several years on open window ventilation in two typical class rooms in the main Engineering Building. Two rooms on the second floor were selected, one with a north exposure and one across the corridor with a south exposure. The building is a 4-story structure with a full ground floor so that the rooms selected were considered to be near mid-height and the stack effect as low as it would be at any place in the building.

Two series of observations have been made, one with the outlet vents closed and the other with the outlet vents open. The building is ventilated by a central fan system. The inlet grilles to the two rooms were kept sealed at all times. The south room was 20 by 30 ft in size and had five windows approximately 3 ft 8 in. by 7 ft 0 in. The outlet grille was 27 in. by 29 in. The north room dimensions were 24 by 26 ft and there were three windows of the same size as in the south room. The outlet grille was also 27 in. by 29 in. Fig. 3 shows a view in the south room, showing an open window and the outlet vent.

The procedure in testing was to open one window in each room a definite height as would be used for open window ventilation. This height was established at 4½ in. An anemometer was set in the center of each window opening and readings observed at 5-min intervals. At the end of one-half hour, anemometers were exchanged between the two rooms and readings continued another one-half hour. This was considered advisable to eliminate any difference in anemometers under variable flow conditions, especially at low velocities. Corrections were applied to the readings, but at very low velocities the readings would not be accurate.

The wind velocity and direction and the outside temperature during the hour of observations were determined from the charts of the local weather station located on the roof of the next building. The charts give a continuous record of the wind velocity and the direction for each minute. Observations were made for winds with a general direction of south and north. In using the results, the one-half hour averages of the velocities through the open window were plotted against the component of the wind that was normal to the building wall. This was obtained by correcting the individual one minute readings of velocity so as to get the component perpendicular to the wall. These values were then averaged for the one-half hour periods.

Fig. 18 shows the results of the tests with the outlet vents open and Fig. 19 shows the results of the tests with the outlet vents sealed. The results for the south winds are shown on the left and for the north winds on the right of these figures. Although the mid-height of the building was chosen to lessen the effect of stack action from temperature differences, temperatures

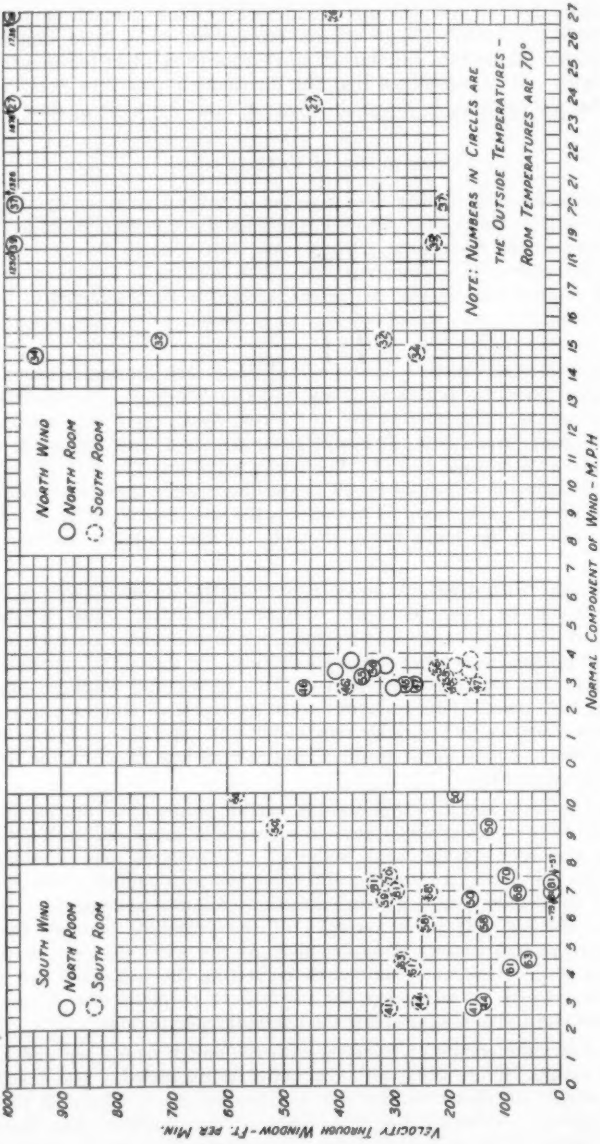


FIG. 18. OPEN WINDOW VENTILATION WITH OUTLET VENTS OPEN

affected the results considerably. The outside temperatures are consequently shown within the circles representing the data points. The north room results are shown as solid circles and the south room results as dotted circles in each case.

The results show that under almost every observed condition there was a considerable inflow of air in both the windward and leeward rooms. Only in one case was an outflow found in even the leeward room and that was with an outside temperature of 81 F and a wind velocity of about 7 mph. The air came into the room through the open window with a velocity ranging from 200 to 500 fpm. The area of the window opening was 1.37 sq ft. In calibrating an anemometer in an opening made to simulate an open window, the coefficient of several runs at different rates of flow was found to be within one per cent of 0.74. The air flow in cfm through the opening used would then be about the same as the velocity in feet per minute.

It is difficult to compare the effect of outlet vents by studying Figs. 18 and 19 because of the effect of outside temperatures. For instance in Fig. 18 for a south wind with outlet vents the points lie lower than in Fig. 19 for the same conditions except without outlet vents. This is due to the higher outside temperatures during the test with outlet vents. At high wind velocities the influence of the outlet vents is noticeable. For instance, with a north wind having a velocity of 21 mph, the velocity of inflow in the windward room according to Fig. 19 would be 975 fpm without outlet vents at an outside temperature of 27 F. Fig. 18 shows that with outlet vents, the inflow was at the rate of 1476 cfm at the same temperature, but a wind velocity of 23.6 mph.

In order to trace the effect of outlet vents on the inflow with open window ventilation an attempt was made to eliminate the temperature effect. The assumptions were made that the wind has no effect on the inflow on the leeward side and that the difference in inflow on the windward and leeward rooms was due to the normal component of the wind velocity. The differences in inflow were found for both north and south winds with and without outlet vents. These differences were then divided by the wind velocity so that a figure was obtained that represented the inflow velocity due to wind effect in feet per minute per mile of wind velocity.

The average found for the windward room with a south wind and with the outlet vents open was 38.8 fpm per mile of wind velocity. The corresponding average for a north wind, the north room now being the windward room, was 40.1 fpm per mile of wind velocity. With the outlet vents sealed the average found with a south wind was 29.8 fpm per mile of wind velocity and with a north wind the average was 24.3 fpm per mile of wind velocity. Most of the tests with the outlet vents sealed were made after a small structure was erected on the north side of the building that sheltered to some extent the test window from northwest winds. In obtaining the average of 24.3 fpm the points with a northwest wind were eliminated and later tests were made only with north and northeast winds. The average with the faulty points included was 21.2 fpm per mile of wind velocity.

The reduction in per cent for the sealing of the outlet vents was 26 for the north wind and 37 for the south wind. This is an average for both of a reduction to 69 per cent for the wind effect on open window ventilation with

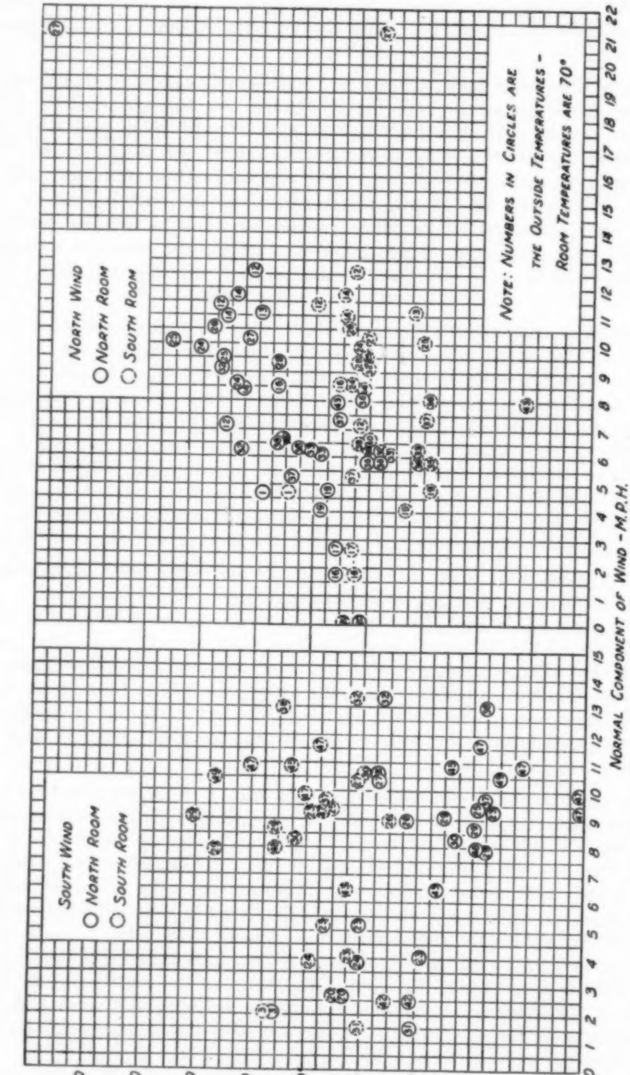


FIG. 19. OPEN WINDOW VENTILATION WITH OUTLET VENTS CLOSED

the elimination of outlet vents. The corresponding figure for the unit ventilator installation in the Mechanical Engineering Building was 75 to 77 per cent.

Fig. 20 shows another study of the effect of outlet vents with open window ventilation in the same two class rooms in the main Engineering Building. Tests were made with the outlet vents open and directly following with the outlet vents closed. In addition to the measuring of the air entering the rooms through the open windows, the air leaving through the outlet vents was measured when they were not sealed.

With a normal wind velocity of 13 mph, 880 cfm entered the windward room and 320 cfm entered the leeward room. In the windward room, 86 per cent as much air left by the outlet vent as entered the open window. In the leeward room, 14 per cent more air left by the outlet vent than entered by the open window. This shows the drift of air from the windward side to the outlet vents of the leeward side. With the sealing of the outlet vents the inflow on the windward side was reduced to 690 cfm. On the leeward side the air flow increased to 360 cfm. The reduction on the windward side amounted to 22 per cent. The only accounting for the increase in inflow on the leeward side of 50 cfm was in the slight reduction of 0.4 mph in wind velocity and the lowering of outside temperature of 3.5 deg.

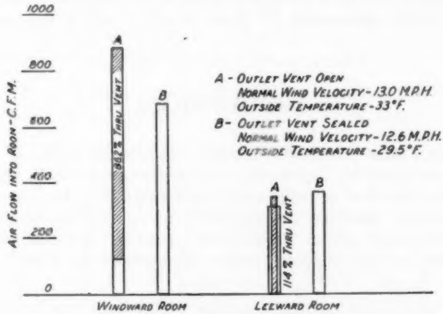


Fig. 20. Outlet Vent Influence on Open Window Ventilation

General Summary

The amount of air passing out through the planned outlet vents of a class room in per cent of the air supply was found to be 55 for a building of comparatively poor construction in so far as tightness to air leakage is concerned. In a new building of very good construction, this per cent was found to be 75. This is not a fixed percentage of the planned air supply but rather is dependent considerably on infiltration. In one series of tests, the air leaving by the planned outlet vent was over 100 per cent of the planned air supply.

The pressures built up were not sufficient to overcome infiltration at ordinary wind velocities. With the planned outlet vent sealed, more than 600 cfm could be supplied to the room by the fan at a pressure equal to that of a 15 mph wind.

Sealing of the planned outlet vents resulted in a reduction in air delivery to the room of from 3 to 25 per cent of the air delivery with the outlet vents open. This percentage depends on the fan characteristics as to the ability to maintain volume of air delivery as the static pressure builds up.

The study of open window ventilation shows an inflow on both the windward and leeward sides of some 200 to 500 fpm under most wind velocities and outside temperatures observed. The inflow was almost 1500 fpm in one case of high wind velocity.

Although rooms were selected near the mid-height of the building, the temperature effect had an influence on the inflow of air through the open windows. In order to obtain a figure for wind effect only, the assumption was made that the difference in inflow in the windward and leeward rooms was due to the normal component of the wind velocity. On this basis, an average of slightly less than 40 fpm resulted from each mile of wind velocity when the outlet vents were open and 27 fpm resulted when the outlet vents were sealed.

DISCUSSION

ALBERT J. NESBITT AND JOHN D. CASSELL (WRITTEN): While the paper is very interesting, there is a good bit of material included therein that is already well known. We refer to standard method of calibrating an anemometer, and the comparison of air distribution for the two systems, and the effect of wind pressure and temperature on the pressure in the rooms tested. We believe that the principal value of the paper lies in the summary of air supplies under different outlet conditions as shown on Fig. 15.

It is especially interesting to note that the outlet vent can be sealed with a drop in capacity of only 8 per cent with a unit system. This would justify the elimination of vent flues insofar as their effect upon the air supply is concerned. One could well argue that it would be good practice to install a unit having a capacity 8 per cent higher than it actually required and then eliminate the vent opening. The economy thus produced would be well worth while since the increase in unit size would be so small as to not materially affect the cost.

The drop off in capacity of 3 per cent for the motor blower indicates the possibility of selecting units with the ability to operate against greater static resistance than that of the unit tested.

Vent openings, however, are not only provided for the purpose of relieving the pressure in the classroom, but in the great majority of school buildings, are used for the definite purpose of directing the air exhausted through the building in wardrobes or cloakrooms.

The outer garments of students are very frequently hung in wardrobes and cloakrooms in a damp condition. It is considered good engineering practice to cause the air from the classroom to be vented through the wardrobe or cloakroom, for the purpose of drying these wraps. The amount of heat given off by this air is frequently sufficient to justify the complete elimination of direct radiation from the cloakroom.

The illustrations shown do not take into account the fact that wardrobes and cloakrooms exist in many of our school buildings. It would, therefore, appear desirable to provide some means of venting from the classroom, if for no other purpose than that of directing air through the spaces where it could do the greatest amount of good rather than letting it leak out under doors or through other openings.

The paper sustains the contention held by us that the amount of area required for a vent opening can very materially be reduced over that generally specified.

In rooms having cloakrooms or wardrobes, it would seem desirable to provide a small vent opening that would maintain a slight pressure in the room, and yet cause about 55 per cent of the air to leave through this opening. The tests further indicate that corridor ventilation is satisfactory, and therefore, these small vent outlets could open into the corridor, thereby eliminating the need for vent flues. In rooms such as illustrated in Figs. 16 and 17, the desired results could be produced by using a corridor door hung about 1 in. off the floor.

The tests further sustain the contention that the unit system with its high velocity, is capable of producing as good distribution, if not better, than the central fan system with a larger air quantity.

It would be interesting to know whether the temperature of the air discharged by the unit ventilator was the same as the temperature of the air supplied by the central fan system. All air circulation is to some extent a function of temperature differences so that even though the air volumes were corrected to 70 deg, a variation in outlet temperatures would affect the distribution results of either system.

The time required for circulation as measured by smoke tests, checks closely with the number of tests we have conducted.

JOHN HOWATT (WRITTEN): Studies of the effect of reducing the area of the air outlets for vents in classrooms in which the air supply is furnished by a central system of fans operating at constant speed were made in a number of Chicago Public Schools in the winter of 1931. The purpose was to determine the effect, if any, on the quality of ventilation and the air distribution in the room by restricting vents. All of the rooms tested had vent ducts leading up to the attic, there connecting with a main vent duct system with roof discharge. The vent ducts from the toilets and kitchens were separated from the ducts from the remainder of the building so were in no way affected by the experiments being conducted.

It was found that invariably as vent openings were restricted, the static pressure in the rooms built up and as it built up the air flow into the rooms through the supply grille decreased. The amount of that decrease varied, however, from room to room and with the location of the room as to whether it was on the leeward or on the windward side of the building, but varied more between different buildings. When the experiments were carried out in a tight, well constructed building with weather stripped windows the air flow into the rooms decreased materially when vents were closed, whereas in the older or more poorly constructed buildings the static pressure built up very little and the decrease of air flow into the rooms was slight. For example, in the tightest constructed building the air flow into the rooms through the supply grille decreased from 1500 cfm to 850 cfm, while in the most loosely constructed buildings it fell only to 1300 cfm per room. Inasmuch as the room vents in these tests were all closed tight the air delivered into the rooms represented the leakage through the cracks around doors and windows and through the walls. The entire building in each case was taken as a test unit, not one room of that building, so the leakage of air from one room to another did not have to be considered. The occupants of the building were satisfied with the ventilation conditions with vent openings closed. There was a noticeable absence of draft and there was no difficulty in keeping the buildings warm by an air blast system of combined heating and ventilation once the buildings had been heated up. There was some complaint, however, that the high air pressures, especially in the tighter buildings caused whistling and rattling noises at doors and windows and odors were noticeable to one coming into the building from outdoors. The practical result of these tests has been a revision in the vent duct sizes in our new building construction.

The studies were so interesting they were carried out through cooperation with the Chicago Board of Health by introducing a measured quantity of carbon dioxide in the rooms and recording the rate of dissipation or dilution of that carbon dioxide. Results were so erratic and unexpected and appeared so much effected by the wind it showed clearly how impossible it must be to predetermine the total value of air that will enter any room through the combination of the mechanical ventilating supply system and infiltration from the outside.

C. J. SCANLAN (WRITTEN): Many questions which have long been in dispute regarding class room ventilation are completely cleared up in this paper. Fig. 20 very clearly presents the performance of open window ventilation, emphasizing the lack of control.

The statement that "the pressures built up were not sufficient to overcome infiltration at ordinary wind velocities" settles the question of how the heat loss calculations for a mechanically ventilated school room should be made. It is believed that this is an important contribution to the art, inasmuch as many engineers have omitted the calculation for infiltration loss when mechanical systems of ventilation were used.

G. E. OTIS AND E. H. BELING (WRITTEN): This well-conceived and carefully executed series of investigations has added considerable knowledge to the field of school classroom ventilation. To the best of our knowledge, this is the first work that has been done on the subject of air outlets in classrooms and we are very hopeful that these investigators will be able to continue their work.

Of particular interest is the fact that in the Dudgeon School the air delivery of the unit ventilator only fell off 8 per cent when the vent outlet was sealed, whereas, in the Mechanical Engineering Building the air delivery of the unit ventilator fell off 23 to 25 per cent with the same conditions. The authors attribute this difference to the fact that the Mechanical Engineering Building is a considerably newer building than the Dudgeon School. Unquestionably, the newness and tightness of the building is at least partly responsible for this difference. However, since these tests were run with two different types of unit ventilators, we wonder if the different fan characteristics of the two unit ventilators was not responsible for at least a part of this difference.

This investigation has brought out in a concrete manner what has long been suspected about central fan systems, namely, that they are sensitive to changes in the individual room conditions. We are hopeful that in future investigations, it will be possible to study the effect of closing the vent outlet in a number of classrooms in buildings of different construction when using the same or similar unit ventilators. We are of the opinion, as a result of our experience in this field, that with a unit ventilator whose characteristic is such that the capacity does not fall off greatly with a slight increase in resistance, the vent outlets may be omitted entirely.

W. C. RANDALL (WRITTEN): I am glad to note that this investigation shows that there is inflow, by natural means, through windows on the leeward side. Objection has been made in the past to the use of windows for ventilation on the basis that windows on the leeward side are only effective for outflow.

The flow through windward windows, according to data given in the text of the paper, is approximately 43, 46, 56 and 44 per cent of the velocity of the wind. That compares favorably with my recommendations of 50 per cent for approximate figuring.

The data shown on Figs. 18 and 19 indicates a variation from the above, particularly for low wind velocities, due probably to temperature effect and the suction effect of the wind at roof outlets. I cannot understand, however, the lower velocities shown in Fig. 18 with outlets open than in Fig. 19, with outlets closed and wonder if, by any chance, the titles or cuts have been reversed.

H. M. Nobis (Written): The following statement is questionable: "It is considered that the distribution was very good in each case and that no evidence is shown in Figs. 16 and 17 for or against planned outlet vents."

As the tests were conducted in unoccupied class rooms they cannot indicate the conditions that exist in occupied rooms.

To warrant the assertion "very good" we should have the benefit of further tests in several occupied classrooms, each one equipped with the various methods of venting. These various venting methods are: *A*. The door vent; *B*. The usual inner wall vent at room center; *C*. The wardrobe vent; *D*. The planned vents consisting of two or more vent outlets, spaced about every eight feet along the inner wall.

The practical results would indicate the *A*. method as medium, *B*. as fair, *C*. as good, and *D*. as very good. In all cases the air would discharge into the adjoining corridor, and the air supply should be large enough to maintain a plus air pressure in the room.

The article itself is a valuable contribution, showing the folly of a central fan installation considered from all practical angles; the effect of room leakage through windows, etc.

Mr. Houghten: In a number of places serious consideration is being given to the question of cutting out stacks in unit ventilated rooms. At the present time the State of Pennsylvania is developing rules to govern the State Board of Education in approving ventilating systems. If such stacks can be eliminated, the result will be a saving. Or, better, perhaps instead of such stacks small amounts of air can be taken through coatrooms or exhausted into the corridors, and a saving will result. We have in Pittsburgh a unit ventilator in one of our offices which happens to be one of the two or three rooms in which O. W. Armspach collected the data on infiltration back in 1920 and 1921. The windows in the room are well weather-stripped and I thought it would be interesting to get a little data in that particular room for which we have infiltration data.

The result corroborates Professor Nelson's paper and shows that with a door of the room open we get a 600 ft velocity. With the door closed and windows closed but no particular caution made to close them tightly, velocity is reduced to 563 ft or a reduction of about 7 per cent. With the doors all closed tightly and some of the larger openings sealed so as to make a very tight room, the reduction is to 549 or 9 per cent.

There is one factor that comes up in connection with cutting out such openings that may be annoying. When you do it a pressure in the room is built up which if it is great enough will cause slamming of corridor doors. I wonder if anything of that sort is noticed in connection with these tests by Professor Nelson.

Mr. Ewald: A comparison between a motor blower and a unit shows that the motor blower does not drop off in capacity nearly as much as the unt. I wish to call attention to the fact that a unit ventilator is essentially a motor blower. You are simply comparing two motor blowers of different fan and motor characteristics.

Authors' Closure: We can vouch for the unpleasantness mentioned by Mr. Houghten in having a very high air pressure in the room. The motor blower which we used was capable of delivering against 6 in. pressure. With the outlet vents sealed, the door and window perimeters sealed, the pressure was around 1 in. It was very difficult with this pressure to open doors and the air whistled under the door causing a very undesirable draft near the floor. Of course, there would be danger of the door slamming.

We agree with Mr. Nesbitt that there are other reasons for having vents to rooms, and the purpose of our paper was not to discourage the use of vents, but just to see

what we do get under various venting conditions. It is possible, as Mr. Houghten says, to reduce the usual vents and still have good distribution and possibly build up the pressure slightly to keep conditions more uniform in the room than we would otherwise have, and to overcome to a certain extent infiltration at fairly low wind velocities.

In the ventilation of cloakrooms, of course, there is a very important reason for taking the air out of the classroom through a planned outlet vent and passing it in the direction in which it may be of further service rather than wasting it by having it go out through window openings as the result of the building up of the pressure in the room.

Mr. Nesbitt asks the question as to the temperature in the room. The system used was a split system, with direct radiation supplying part of the heat, and we noticed no difference in temperature in various parts of the room in any case. Figs. 16 and 17 show distribution tests that we made and it seemed to us that the circulation of air was quite active with or without planned outlet vents. There was little temperature difference from one place to another whether the outlet vents were used or not used.

In the tests with the crack open below the door, which was half an inch wide, that crack acted as an outlet vent and attracted the air to the same place it would be attracted if the louver opening had been open.

Mr. Randall has called attention to the fact that Fig. 18 shows lower velocities with outlet vents open than Fig. 19 shows with outlet vents closed. This is explained in the paper in connection with these figures as being due to the higher outside temperatures during the test with outlet vents. At high wind velocities the influence of the outlet vents is more pronounced and the influence of outside temperature differences is less noticeable.

Mr. Ewald mentioned the question of motor blowers. It is true that the unit ventilator is a motor blower as well as the one we called the motor blower, but in one case the fan was capable of delivering against a six-inch pressure whereas in the other case it was capable of delivering reasonable air volumes at very low pressures only; so that they are quite different and we made the study of the two just to get the difference in the results.

If it were desirable not to use outlet vents, of course, a unit ventilator could be built to work against a considerable pressure, but at an increased cost of operation and a probable increase in noise.

No. 937

THERMAL PROPERTIES OF BUILDING MATERIALS

By F. B. Rowley[1] and A. B. Algren,[2] Minneapolis, Minn.

(*MEMBERS*)

This paper is the result of research conducted at the University of Minnesota in cooperation with the A. S. H. V. E. Research Laboratory

IN THE cooperative research work between the American Society of Heating and Ventilating Engineers and the University of Minnesota, one of the problems has been that of analyzing and studying the laws governing heat flow through insulating materials and built-up wall constructions. The various parts of the problem have consisted in determining (1) For average walls, suitable coefficients for the surfaces for homogeneous materials used in their construction, and for air spaces, (2) By test methods the overall heat transmission coefficient for different types of wall constructions, and (3) Overall heat transmission coefficients by calculation and comparing these results with those obtained by test methods.

Reports covering the details of the apparatus used and much of the test data have been published.[3]

Since these reports were published, test data have been obtained on several types of walls, including tile, brick, concrete, masonry, and some others of special design. The object of this paper is to report those data and test results which have not heretofore been published. For details of apparatus, method of procedure, etc., reference should be made to the previous papers.

---

[1] Professor of Mechanical Engineering and Director of Experimental Engineering Laboratory, University of Minnesota.

[2] Instructor in Mechanical Engineering and Assistant Director of Experimental Engineering Laboratory, University of Minnesota.

[3] Heat Transmission Research, by F. B. Rowley, F. A. Morris and A. B. Algren (A. S. H. V. E. Transactions, Vol. 34, 1928). Overall Heat Transmission Coefficients Obtained by Tests and by Calculation, by F. B. Rowley, A. B. Algren and J. L. Blackshaw (A. S. H. V. E. Transactions, Vol. 35, 1929). Thermal Resistance of Air Spaces, by F. B. Rowley and A. B. Algren (A. S. H. V. E. Transactions, Vol. 35, 1929). Effects of Air Velocities on Surface Coefficients, by F. B. Rowley, A. B. Algren and J. L. Blackshaw (A. S. H. V. E. Transactions, Vol. 36, 1930). Surface Conductances as Affected by Air Velocity, Temperature, and Character of Surface, by F. B. Rowley, A. B. Algren and J. L. Blackshaw (A. S. H. V. E. Transactions, Vol. 36, 1930). Surface Coefficients as Affected by Direction of Wind, by F. B. Rowley and William A. Eckley (A. S. H. V. E. Transactions, Vol. 38, 1932.).

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Milwaukee, Wis., June, 1932, by A. B. Algren.

DESCRIPTION OF WALLS TESTED

The construction of the various walls tested by the hot box method to determine the overall coefficients is shown in detail in Figs. 1 to 9, inclusive, and a summary of test results is given in Table 1. In all cases where building

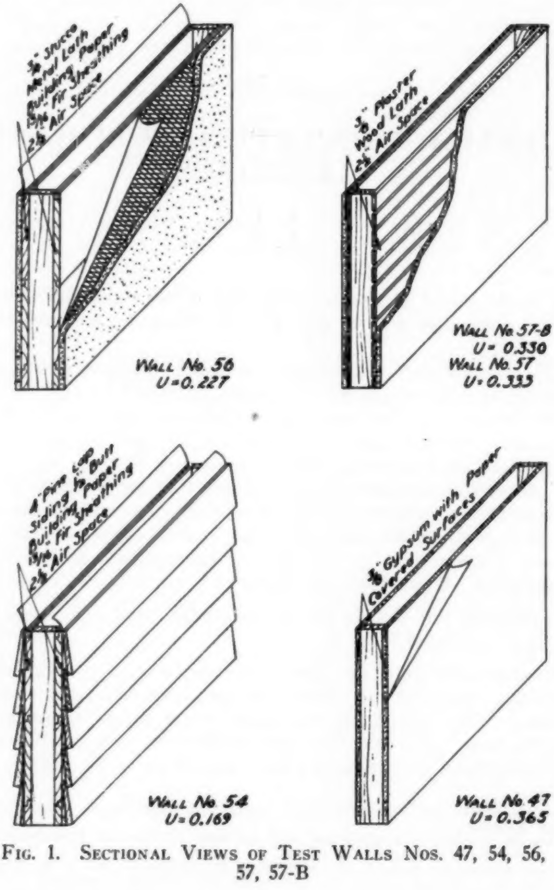


FIG. 1. SECTIONAL VIEWS OF TEST WALLS NOS. 47, 54, 56, 57, 57-B

paper was used in the walls, it was that known as No. 2 Building Felt, with an average weight of 44 to 55 lb per roll of 324 sq ft. In all walls where plastered surfaces or mortar joints were used, the wall was built and allowed to season a sufficient length of time before the test was made to make sure that the mortar or plaster was thoroughly dried out. In the case of concrete

walls, the tests were made at various periods in the age of the wall as indicated in the discussion of the results for these tests.

DISCUSSION OF RESULTS

The coefficient of heat transmission U shown with the figure giving the

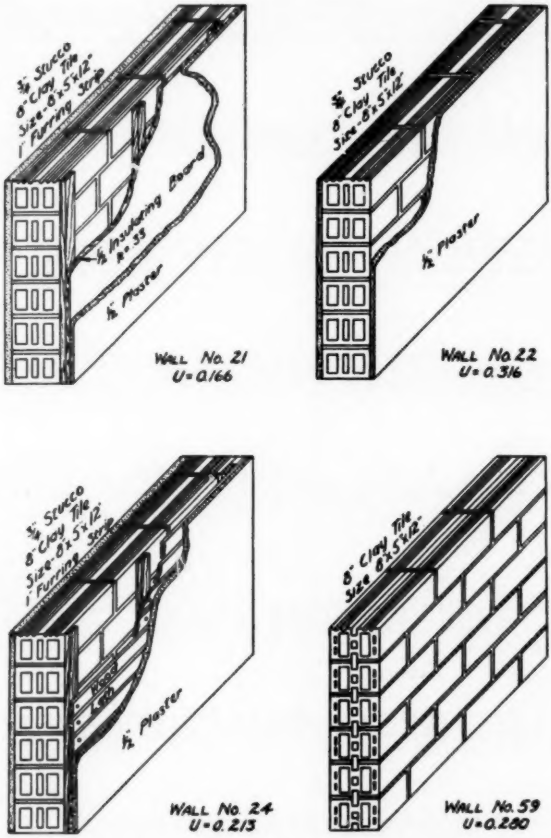


FIG. 2. SECTIONAL VIEWS OF TEST WALLS NOS. 21, 22, 24, 59

detailed construction of the wall is that obtained by test for still air conditions. Table 1 shows this coefficient and, also, a transmission coefficient corrected for a wind velocity over the outside surface of the wall of 15 mph. In making this correction, an average outside surface coefficient of 1.65 was taken as that obtained by the test on a particular wall. This was the average

TABLE 1. SUMMARY OF RESULTS OF TESTS ON WALL CONSTRUCTION BY THE HOT BOX METHOD

Date of Test	Wall No.	Test No.	Air Temperatures, Deg Fahr			Surface Constants		Coefficient of Transmission U	
			High Side	Low Side	Mean Temp.	Inside f_i	Outside f_o	By Test	Corrected to a Wind Velocity of 15 mph
2/17/28	21	431	80.5	0.	40.2	1.844	1.403	0.167	0.181
2/21/28	21	432	80.6	0.	40.3	1.830	1.629	0.166	0.180
3/21/28	22	445	80.05	0.05	40.1	2.007	2.188	0.318	0.375
3/22/28	22	446	80.2	0.	40.1	1.981	2.168	0.314	0.369
3/28/28	24	448	80.2	—0.05	40.1	1.781	1.651	0.212	0.232
3/29/28	24	449	80.2	0.	40.1	1.789	1.847	0.214	0.234
9/1/28	37	487	80.3	—0.15	40.07	1.675	0.859	0.101	0.106
9/27/28	37	488	80.25	—0.35	39.95	1.819	0.914	0.107	0.113
10/2/28	37	490	80.37	—0.23	40.07	1.834	0.720	0.099	0.104
11/23/28	37	512	81.48	—1.0	40.24	1.855	1.291	0.106	0.111
10/30/28	42	502	80.37	0.	40.18	1.658	1.235	0.135	0.144
10/30/28	42	502A	80.39	0.	40.19	1.678	1.465	0.138	0.147
12/4/28	47	516	80.26	—0.03	40.11	1.667	1.448	0.365	0.427
12/27/28	50	525	79.81	—0.57	39.62	1.680	1.567	0.499	0.640
2/5/29	50	541	80.46	—0.29	40.08	1.662	1.624	0.499	0.640
1/29/29	51	539	80.67	—0.38	40.14	1.690	1.548	0.357	0.425
4/23/29	51	566	80.59	—0.37	40.11	1.714	1.591	0.354	0.422
2/1/29	52	540	80.39	—0.32	40.03	1.808	1.571	0.412	0.504
4/18/29	52	565	80.66	—0.57	40.04	1.841	1.619	0.401	0.487
1/10/29	54	532	80.33	—0.51	39.91	1.625	1.524	0.171	0.183
4/30/29	54	568	80.79	—0.98	39.90	1.716	1.526	0.168	0.180
1/25/29	55	538	80.81	—0.48	40.1[illegible]	1.646	1.668	0.588	0.792
4/26/29	55	567	80.90	—1.07	39.91	1.646	1.625	0.572	0.762
11/5/29	55	583	80.27	—0.02	40.12	1.665	1.605	0.579	0.776
9/23/30	55	597	79.5	0.40	39.95	2.615	1.721	0.582	0.780
2/7/29	56	542	79.79	—0.04	39.88	1.768	1.486	0.227	0.254
2/8/29	56	543	79.81	0.0	90.90	1.777	1.493	0.227	0.254
2/14/29	57	544	79.73	—0.16	39.78	1.700	1.585	0.331	0.381
2/15/29	57	545	79.74	0.05	39.89	1.716	1.677	0.335	0.387
5/10/29	57B	573	79.47	—0.33	39.57	1.649	1.511	0.330	0.380
6/18/29	59	575	80.94	—0.90	40.02	1.747	1.505	0.280	0.319
6/20/29	60	576	81.24	—1.08	40.08	1.831	1.474	0.266	0.301
6/22/29	61	577	79.93	0.02	39.98	1.602	1.561	0.270	0.306

(Note 1) Average surface conductance (f_o) for walls as tested = 1.650.
(Note 2) Surface conductance (f_o) for pine surface for outside wind exposure of 15 mph = 4.75.
(Note 3) Surface conductance (f_o) for stucco surface for outside wind exposure of 15 mph = 7.50.

(Note 1) Average surface conductance (f_o) for walls as tested = 1.650.
(Note 2) Surface conductance (f_o) for pine surface for outside wind exposure of 15 mph = 4.75.
(Note 3) Surface conductance (f_o) for stucco surface for outside wind exposure of 15 mph = 7.50.

TABLE 1 (*continued*). SUMMARY OF RESULTS OF TESTS ON WALL CONSTRUCTION BY THE HOT BOX METHOD

Date of Test	Wall No.	Test No.	Air Temperatures, Deg Fahr			Surface Constants		Coefficient of Transmission U	
			High Side	Low Side	Mean Temp.	Inside f_i	Outside f_o	By Test	Corrected to a Wind Velocity of 15 mph
7/9/29	61A	582	79.98	0.13	40.05	1.777	1.480	0.265	0.300
6/27/29	62	579	79.92	0.01	39.96	1.815	1.884	0.205	0.226
7/2/29	63	580	79.93	0.22	40.07	1.726	1.800	0.343	0.403
6/25/29	64	578	79.94	0.14	40.04	1.708	1.318	0.265	0.299
7/6/29	65	581	79.99	0.13	40.06	1.733	1.528	0.255	0.290
5/24/30	66	589	80.30	-0.05	40.12	1.711	1.627	0.310	0.360
5/27/30	67	590	80.40	-0.15	40.12	1.732	1.654	0.320	0.372
10/2/30	68	599	79.81	0.05	39.93	1.803	2.247	0.590	0.795
9/26/30	69	598	79.89	0.09	39.99	1.924	1.655	0.585	0.785
6/10/30	70	594	80.09	0.10	40.09	1.544	1.981	0.358	0.425
10/9/30	70	601	79.80	0.20	40.00	1.618	1.618	0.353	0.418
6/13/30	71	595	80.10	0.15	40.12	1.594	1.605	0.460	0.576
10/7/30	71	600	79.85	0.10	39.97	1.662	1.559	0.457	0.571
6/6/30	72	593	80.07	0.0	40.04	1.571	1.575	0.436	0.540
6/3/30	73	592	79.85	0.35	40.10	1.632	1.648	0.408	0.497
6/18/30	74	596	80.08	0.33	40.20	1.604	1.895	0.413	0.504
1/20/31	75	627	79.98	0.13	40.05	1.623	1.477	0.540	0.707
1/14/31	76	626	79.90	0.04	39.97	1.622	1.529	0.553	0.729
10/15/30	78	602	79.82	0.52	40.17	1.954	1.616	0.691	.....
2/5/31	78B	631	80.22	-0.43	39.89	1.246	0.923	0.460	.....
10/27/30	79	606	78.89	-0.04	39.42	1.761	1.588	0.264	.....
10/30/30	79A	608	80.16	-0.08	40.04	1.280	1.164	0.169	.....
11/10/30	79B	610	80.12	0.06	40.09	1.230	1.467	0.175	.....
11/18/30	79B	611	79.97	0.02	39.99	1.252	1.291	0.171	.....
10/28/30	80	607	79.87	-0.27	39.80	1.746	1.407	0.257	.....
5/26/31	80A	661	81.00	2.08	41.54	0.814	1.757	0.130	.....
1/6/31	81	624	79.06	-0.20	39.43	1.671	1.542	0.344	0.405
1/8/31	82	625	80.02	0.	40.01	1.701	1.429	0.379	0.454
11/25/30	83	613	79.60	0.07	39.83	1.726	1.623	0.407	0.495
4/13/31	90	650	80.38	-0.17	40.10	1.661	1.533	0.344	0.405
4/16/31	91	651	80.27	-0.01	40.13	1.719	1.532	0.354	0.418
4/20/31	92	652	80.31	0.19	40.25	1.760	1.575	0.337	0.396

(Note 4) Surface conductance (f_o) for brick surface for outside wind exposure of 15 mph = 6.10.
(Note 5) Surface conductance (f_o) for concrete, tile, cinder block, and rubble stone surfaces for outside wind exposure of 15 mph = 5.78.
(Note 6) Surface conductance (f_o) for plaster surface for outside wind exposure of 15 mph = 4.68.

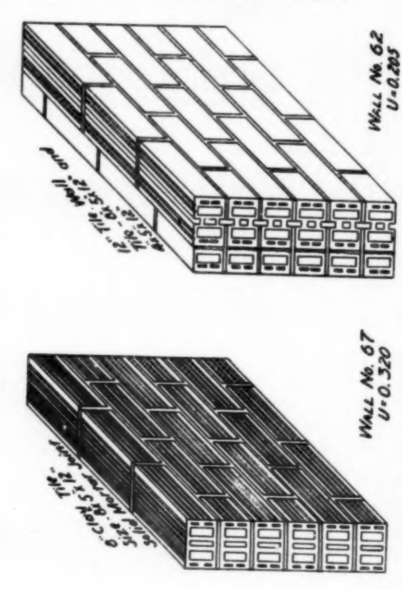


FIG. 4. SECTIONAL VIEWS OF TEST WALLS NOS. 61, 62, 65, 67

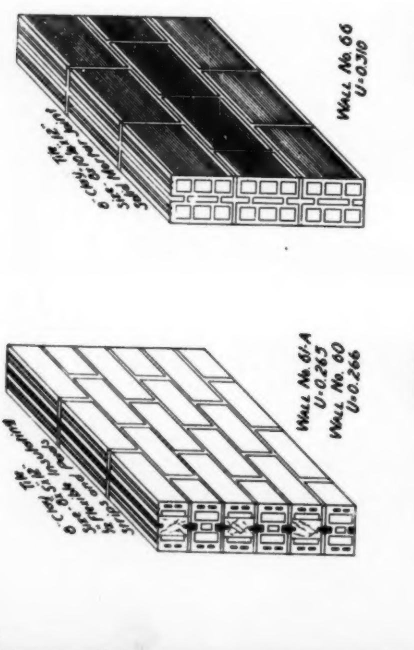


FIG. 3. SECTIONAL VIEWS OF TEST WALLS NOS. 60, 61-A, 63, 64, 66

coefficient for all tests which were made under the same conditions of air velocity over the cold surface of the wall, and was used on account of the difficulty of obtaining exact surface coefficients for several of the rough surface walls. The corrected coefficients for a wind velocity of 15 mph were taken from test data previously published for surface coefficients. In order to make them applicable to average conditions, the values obtained for parallel air flow were reduced by 15 per cent. This corresponds to the average reduction for different angles of flow as shown by the curves given in the paper, Surface Coefficients as Affected by Direction of Wind.[3]

The following formula was used in changing the coefficients of transmission from the test condition to that for a wind velocity over the outside surface of 15 mph:

$$\frac{1}{U_x} = \frac{1}{U} - \frac{1}{f_1} + \frac{1}{f_x} \tag{1}$$

where

U = coefficient of transmission for test

U_x = coefficient of transmission for a wind velocity over the outside surface of 15 mph

f_1 = outside surface conductance for test condition

f_x = outside surface conductance for a 15-mile wind velocity

Substituting values for Wall 21, Table 1 in Formula 1:

$$\frac{1}{U_x} = \frac{1}{0.167} - \frac{1}{1.650} + \frac{1}{7.5} = 5.505$$

$$U_x = 0.181$$

Values used for surface conductances for the surfaces tested are shown in the footnote of Table 1.

Fig. 1 represents four walls which were built up, using the same surface finish on each side of an air space partition in order to determine the thermal properties of the particular combination of materials as commonly used in finishing the surfaces of frame walls. For Wall 56, both surfaces were covered with 13/16 in. fir sheathing, building paper, metal lath, and ¾-in. stucco. For Wall 57 and Wall 57B, both surfaces were covered with ⅜-in. wood lath and ⅜-in. gypsum plaster. For Wall 54, both surfaces were covered with 13/16-in. fir sheathing, building paper and 4-in. pine lap siding, and for Wall 47 both surfaces were covered with ⅜-in. gypsum board papered on both surfaces. It will be noted that Walls 57 and 57B are of the same construction and tested under the same conditions. In several cases, duplicate walls were made in order to determine the differences to be expected in construction. In all such cases, the differences in test values were very small, although it should be remembered that in practice very great differences in workmanship are possible.

[3] Heat Transmission Research, by F. B. Rowley, F. A. Morris and A. B. Algren (A. S. H. V. E. TRANSACTIONS, Vol. 34, 1928). Overall Heat Transmission Coefficients Obtained by Tests and by Calculation, by F. B. Rowley, A. B. Algren and J. L. Blackshaw (A. S. H. V. E. TRANSACTIONS, Vol. 35, 1929). Thermal Resistance of Air Spaces, by F. B. Rowley and A. B. Algren (A. S. H. V. E. TRANSACTIONS, Vol. 35, 1929). Effects of Air Velocities on Surface Coefficients, by F. B. Rowley, A. B. Algren and J. L. Blackshaw (A. S. H. V. E. TRANSACTIONS, Vol. 36, 1930). Surface Conductances as Affected by Air Velocity, Temperature, and Character of Surface, by F. B. Rowley, A. B. Algren and J. L. Blackshaw (A. S. H. V. E. TRANSACTIONS, Vol. 36, 1930). Surface Coefficients as Affected by Direction of Wind, by F. B. Rowley and William A. Eckley (A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932).

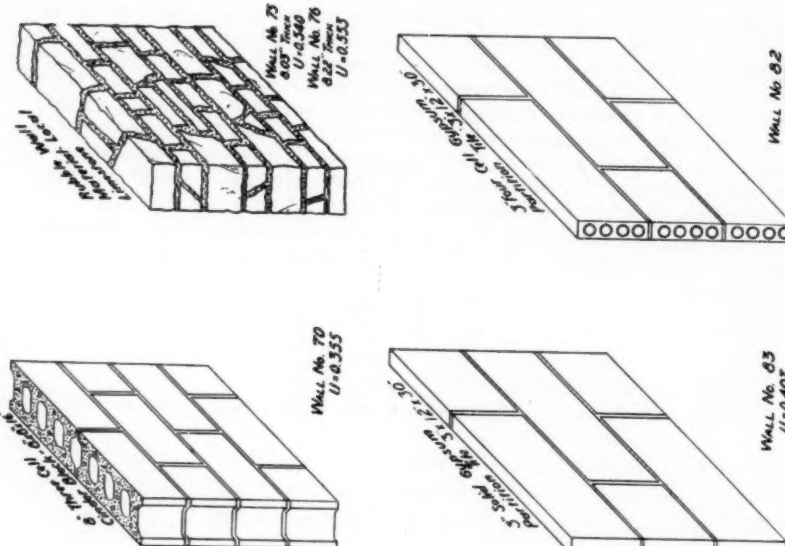


FIG. 6. SECTIONAL VIEWS OF TEST WALLS NOS. 70, 75, 76, 82, 83

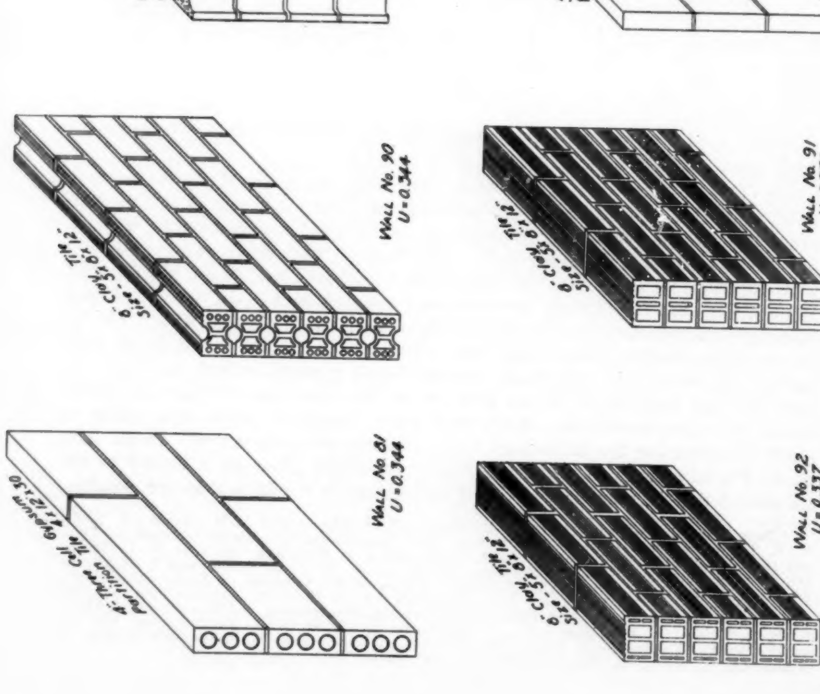


FIG. 5. SECTIONAL VIEWS OF TEST WALLS NOS. 81, 90, 91, 92

The coefficients for these walls are given at a mean temperature of 40 F, although they were actually obtained for mean temperatures varying from 30 F to 110 F. The results for these tests are shown by the curves of Fig. 10.

CLAY TILE WALLS, TEST RESULTS

Figs. 2, 3, 4, and 5 show a group of 15 clay tile walls. Fourteen of these walls are 8 in. thick, are built with tile of different cell structure, and are treated differently as to joints and surface finish. The range in values of the conductivity coefficient *U* for the unfinished walls is from 0.354 for Wall 91, Fig. 5, to 0.265 for Wall 64, Fig. 3.

Wall 63, Fig. 3, and Wall 91, Fig. 5, are built of a common type of 3-cell tile, the construction being similar, although the tiles were obtained from different manufacturers. The coefficients for these walls are 0.343 and 0.354, respectively, which may be taken as reasonable averages for this type of construction.

In Walls 92 and 67, the air cells have been broken up in such a manner as to give substantially 4 cells for Wall 92 and 5 cells for Wall 67. The coefficients of conductivity are 0.337 and 0.320, respectively, indicating that, as the air cells are broken up into greater numbers, the conductivity of the finished wall is decreased.

In each of the 4 walls, Nos. 63, 91, 92, and 67, there is a direct path for the flow of heat through the top and bottom surfaces of the individual tile. In other words, there is a direct line of low heat-resisting material through which heat may flow from the hot to the cold surface of the wall. As a direct contrast to this, consider Wall 64 in which the air spaces are staggered, giving a longer path through the material for the flow of heat from surface to surface. In the construction of this wall, care was exercised to keep the center opening free from mortar. The coefficient *U* is 0.265, which is a material reduction over the other type.

Wall 66 was built of the same tile as Wall 64, but, in this case, the joints were filled solid with mortar. The coefficient was increased to 0.31 as compared with 0.265 when the joints were not filled. Wall 65 was built of the same tile as Wall 64, but, for Wall 65, $\frac{1}{2}$-in. insulating strips were placed horizontally in the open air spaces at the joints, and insulating pads were placed at the end of each tile as shown in Fig. 4. These insulating pads and strips gave slightly better insulation than the air space and prevented any mortar from getting into the joints, and, as shown from the results, reduced the coefficient to 0.225 as compared with 0.265 without the strips. This reduction, however, may have been partly due to the surface finish on the wall.

Walls 59, 60, and 61A were built of the same tile, the difference being that Walls 60 and 61A have insulating strips placed in the joints, filling the central air space between the tile, and, also, insulating pads at the end of the tile. In these walls, the insulating strips reduced the coefficient from 0.28 to 0.265. Wall 61 was of the same construction as Walls 60 and 61A, with the exception that the $\frac{3}{8}$-in. plaster was applied to the inner surface of Wall 61. The coefficient for the plastered wall was 0.27 as compared with 0.265 for the wall without the plaster. This increase may have been due to some slight difference in surface coefficients or to a difference in construction which can

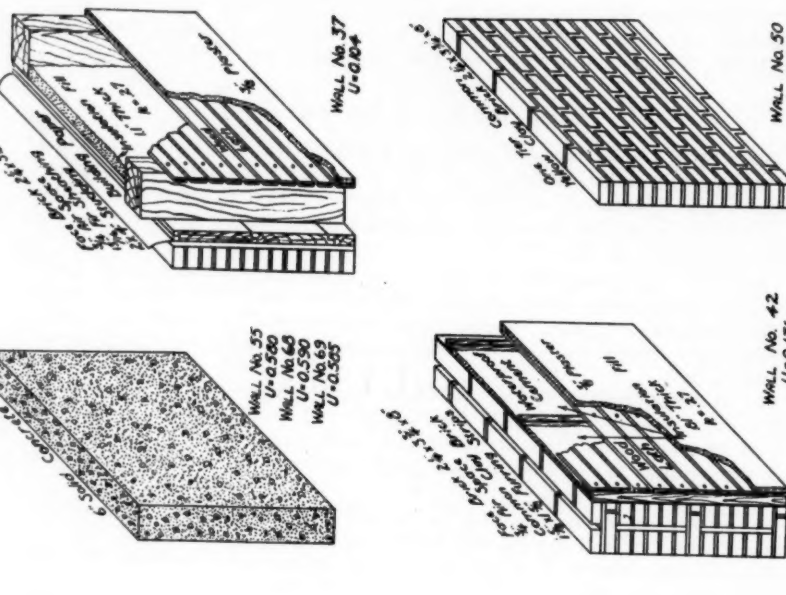


FIG. 8. SECTIONAL VIEWS OF TEST WALLS Nos. 37, 42, 50, 55, 68, 69

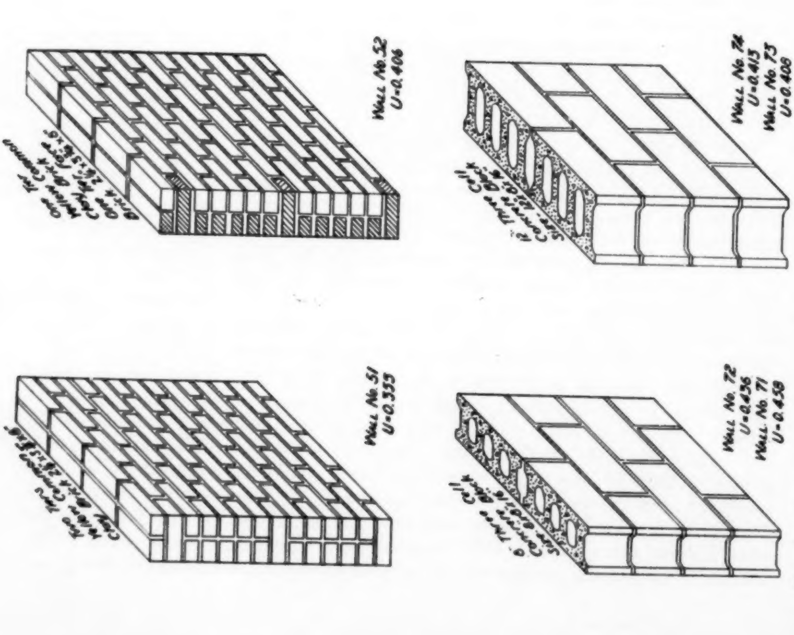


FIG. 7. SECTIONAL VIEWS OF TEST WALLS Nos. 51, 52, 71, 72, 73, 74

always be expected in walls of this character. Wall 62 was built of the same tile as Wall 59, the difference being that it was built 12 in. wide, using one and one-half thicknesses of the tile. If for these walls the surface coefficients are deducted, it is found that the conductances are proportional to the thicknesses.

In general, it may be said that the efficiency of a tile wall is increased by breaking up the air spaces and by eliminating any direct paths or tile connections between the two surfaces of the wall. The greatest improvement seems to be possible by breaking up the direct path for heat flow.

The effect of different surface finishes on the tile walls is shown by Walls 21, 22, and 24 of Fig. 2. In these walls the same tile was used, the exterior finish in each case being ¾ in. stucco and the interior finish being ½ in. plaster applied direct to the tile for Wall 22; applied to wood lath furred out for Wall 24, and applied to ½-in. insulating board furred out for Wall 21. Wood lath on 1-in. furring strips reduces the coefficient from 0.316 to 0.213, while ½-in. insulating board, with a coefficient of conductivity k of 0.33, on 1-in. furring strips reduces the coefficient to 0.167.

CONCRETE WALLS, TEST RESULTS

Concrete walls, Nos. 55, 68, and 69, Fig. 8, were of monolithic construction, Nos. 71 and 72, Fig. 7, were built of 8-in. 3-cell concrete blocks, and Nos. 73

TABLE 2. EFFECT OF AGE ON CONDUCTIVITY OF CONCRETE

Wall No.	Age of Wall	Thermal Conductivity k
55	1 month 5 days	12.3
55	4 months 6 days	11.2
55	11 months 15 days	11.6
55	22 months 3 days	11.8
55	36 months	10.7
68	10 months	12.4
68	26 months 21 days	11.6
69	9 months 17 days	12.1

and 74, Fig. 7, were built of 12-in. 3-cell concrete blocks. Referring to the first 3, No. 55 was built of a 1:2:4 mix and Nos. 68 and 69 were built of a 1:2½:4 mix. Wall 68 was mixed with water to give a 6-in. slump, and Wall 69 was mixed to give a 3-in. slump. In all cases, No. 4 sand was used, that is sand of which 95 per cent passed through a No. 4 sieve, and the gravel was graded from 1½ in. to ¼ in. The results of these three tests were so close that it would indicate that the slump tests had nothing to do with the thermal conductivity of the material.

In concrete walls, it is some time before the moisture is thoroughly eliminated and the conductivity coefficient becomes uniform. In order to get the effect of time on the conductivity of the concrete, tests were made at different ages, as shown in Table 2. The results of these tests indicate that the coefficient of conductivity k for concrete as tested is about 11.5 after the concrete has been thoroughly cured. This coefficient will naturally be somewhat different for different aggregates and mixes.

The two 8-in. concrete block walls, Nos. 71 and 72, were built of blocks which were of the same dimensions but which were purchased from different manufacturers. The variation in these test results from 0.436 to 0.458 may be accounted for partly by the possible difference in the amount of mortar placed in the joints, partly by a difference in the aggregate used, and possibly, to experimental differences. The 12-in. concrete block walls Nos. 73 and 74 were likewise built from blocks of the same dimensions but purchased from different manufacturers. In this case, the test results, 0.408 and 0.413, show a very close agreement.

Wall 70, Fig. 6, was built from 8-in. 3-cell cinder concrete blocks purchased on the open market. In these blocks, the aggregate was made completely of cinders, and the density was found to be 105 lb per cubic foot. There is naturally a variation in cinders, and any one test cannot be con-

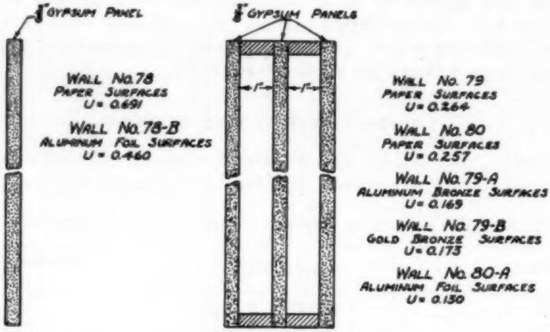


FIG. 9. WALLS CONSTRUCTED TO SHOW THE EFFECT OF SURFACE FINISH ON HEAT TRANSMISSION

sidered as a standard. However, the coefficient of 0.355 as compared with the coefficient of 0.447 for walls of the same type of concrete block indicates a considerable reduction in heat transmission for cinder-concrete.

BRICK WALLS, TEST RESULTS

In Figs. 7 and 8, five brick walls are shown which are constructed either entirely of brick or of brick in combination with other materials. Walls 50 and 51 are both constructed of common yellow bricks, 8 in. x 3¾ in. x 2½ in. in dimension. Wall 50 is a single row and Wall 51 a double row of brick. If for these walls the inside and outside surface coefficient be taken as 1.65, the thermal conductivity $k = 5.0$.

Wall 52 is an 8-in. wall composed of one layer of 4-in. common yellow brick and one layer of pressed face brick. The coefficient U for this wall is 0.406 as compared with 0.356 for the same thickness of wall built entirely of common brick, indicating that the surface brick has a much higher coefficient of conductivity than the common yellow clay brick. Wall 37 shows an insulated frame wall with a brick veneer finish, and Wall 42 shows an 8-in. brick wall with insulating material, furring lath, and plaster on the inside.

GYPSUM PARTITION TILE WALLS, TEST RESULTS

Wall 81 of Fig. 5 and Walls 82 and 83 of Fig. 6 were built of different thicknesses of gypsum partition tile as shown. By comparing the results for Walls 82 and 83, it is found that the cylindrical openings through the tile reduce the conductivity coefficient for the 3-in. wall from 0.407 to 0.379, or, in other words, these openings introduce a heat resistance into the wall equal to 0.18. If for Wall 83 the average inside and outside surface coefficients are taken to be 1.65, the thermal conductivity per inch of material is found to be 2.5.

RUBBLE WALLS, TEST RESULTS

Walls 75 and 76, Fig. 6, were built of limestone and were approximately 8 in. thick. The conductivities of these walls were 0.54 and 0.553, respectively, a variation well within experimental limits for walls of this type. If an average overall coefficient is used and the surface coefficients are taken as 1.65 for both inside and outside surfaces, the thermal conductivity k for the wall is found to be 12.5, or slightly higher than that found for concrete walls. It is probable, however, that the surface coefficients may be somewhat higher than the average of 1.65, due to the uneven character of the surfaces.

TABLE 3. COMPARATIVE HEAT RESISTANCES FOR SURFACES AND AIR SPACES LINED WITH PAPER, BRONZE PAINT, AND ALUMINUM FOIL

Wall No.	Coefficient of Transmission U	Heat Resistance $\frac{1}{U}$	Surface Resistances $\frac{1}{f_o}+\frac{1}{f_i}$ Calculated	Heat Resistance of Material	Resistance of 1-in. Air Space Calculated
78	0.691	1.446	1.212	0.234	....
78B	0.460	2.172	1.938	0.234	....
79	0.264	3.780	1.212	0.702	0.93
80	0.257	3.890	1.212	0.702	0.93
79A	0.169	5.920	1.639	0.702	1.79
79B	0.171	5.840	1.573	0.702	1.78
80A	0.130	7.690	1.938	0.702	2.52

EFFECT OF MEAN TEMPERATURE ON HEAT TRANSMISSION THROUGH BUILT-UP WALL SECTIONS

In built-up wall sections and insulating materials, there is a variation in the heat transmission coefficients with variations in mean temperatures. The coefficient is increased as the mean temperature through the test section is increased, and, for practical purposes, this increase is a straight-line relation.

Fig. 10 shows a set of curves giving the test results for walls at mean temperatures varying from 30 F to 110 F. With the exception of Wall 80A, the surfaces of which were covered with aluminum foil, the variation is substantially the same for all walls. This variation is in general somewhat greater than that found for homogeneous materials, probably due to the more definite air spaces in the construction of the walls.

SURFACE EFFECT ON HEAT TRANSMISSION

Heat is transmitted from the surface of a material by convection, conduction, and radiation. The amount given off by convection is governed by the roughness of the surface and wind velocity. The effect of wind velocity on average building surfaces has been shown in the results previously published.

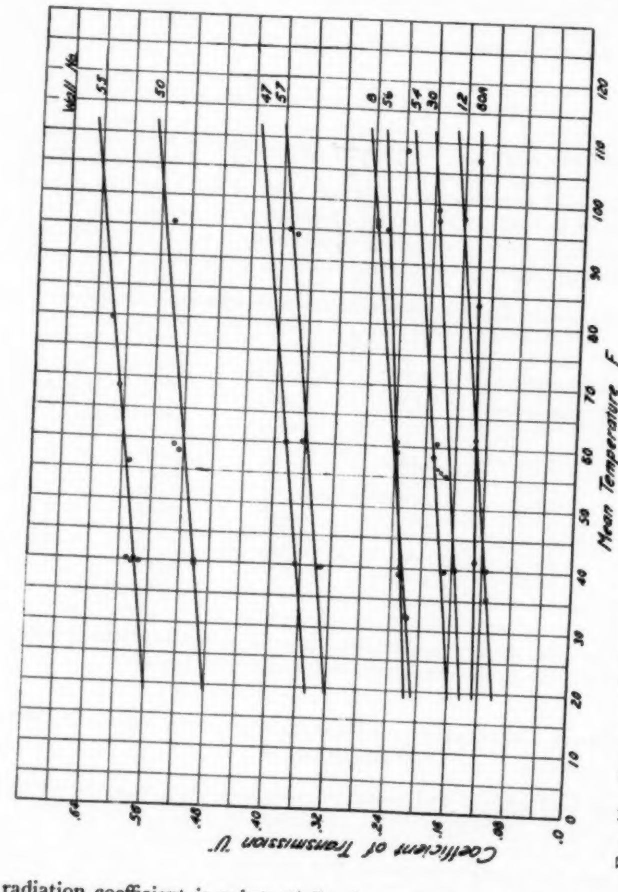


FIG. 10. RELATION BETWEEN COEFFICIENT OF HEAT TRANSMISSION U AND MEAN TEMPERATURE FOR VARIOUS TYPES OF WALL CONSTRUCTION

The radiation coefficient is substantially the same for such materials as paper, wood fiber materials, plaster, concrete surfaces, glass, and, in fact, the majority of building materials. Certain metal surfaces, however, have a greatly reduced radiation coefficient. In order to show the effect of a few of these surfaces, several walls were constructed of ⅜-in. gypsum board, part of them with air spaces and part of them without air spaces. The surfaces of the boards

were covered with paper, aluminum bronze, gold bronze, and aluminum foil. The details of the construction of the several walls are shown in Fig. 9.

These walls were tested by the hot box method, the results of the tests being given in Table 1 and in Table 2. Referring to Table 3, the coefficients of heat transmission *U* are the test results as determined at a mean temperature of 40 F. The over-all resistance was taken as the reciprocal of the coefficient *U*. The surface resistances were calculated by taking an average surface coefficient for the paper-covered wall of 1.65. This applied directly to Walls 78, 79, and 80. The surface resistance for Wall 78B was taken as the difference between the over-all heat resistance and the resistance of the material in the wall. The surface resistances for Walls 79A and 79B were taken from test data and the surface resistances for Wall 80A were taken the same as those for Wall 78B, since the surface covering was of the same material. The resistance of the gypsum board from which the walls were made was taken from test data for Wall 78, using 3 times this value for those walls in which three sheets of material were used.

The heat resistance of the air spaces shown in the last column of Table 3 was calculated from the over-all resistance, the surface resistances, and the resistance of the solid material of the wall. The resistance for Walls 79 and 80, in which the surfaces were lined with paper, checked almost exactly with the resistance as previously determined for 1-in, air spaces lined with fibrous material. In the previous work, the resistance for 1 in. air space at a mean temperature of 40 F was found to be 0.90, whereas, in these tests it was found to be 0.93. This would indicate that the method used in calculating the air space resistances gives reasonably accurate results. The resistances for the various parts of Walls 79A and 79B, in which the surfaces are covered with aluminum bronze and gold bronze, respectively, are substantially the same. The resistance of the air space bounded by aluminum foil is shown as 2.52, or nearly three times that of the air space bounded by paper. These tests showed decidedly the importance of radiant heat as a factor in surface transmission even at the relatively low temperatures as used in building construction.

DISCUSSION

PAUL D. CLOSE (WRITTEN): It is interesting to compare the results reported in this paper with the computed heat transmission coefficients in THE GUIDE. Unfortunately many of the types of construction in the paper are not identical with those in THE GUIDE, thus making an accurate comparison difficult.

Hollow Tile Walls: This is particularly true of the hollow tile walls, of which material there are many varieties. The hollow tile walls in THE GUIDE were based on types of this material commonly used in the east. The conductances for the tile which are given in Table 8, Chapter 3, THE GUIDE 1932, were estimated from air space conductances reported in the paper, Thermal Resistance of Air Spaces, by F. B. Rowley and A. B. Algren (A.S.H.V.E. TRANSACTIONS, Vol 35, 1929), whereas the conductivity of the structural material of the tile or burned clay was arbitrarily assumed to be the same as the value used for brick, namely, 5.0 per 1 in. If the air space values are reasonably correct—and there is no reason to doubt their accuracy—the effect of any error which may have resulted from the assumption of a conductivity of 5.0 per 1 in. for the burned clay would be minimized by the rela-

tively large proportion of the total resistance of the tile due to the air spaces. Therefore, it seems logical to assume that in the case of walls of this type, the number of air cells in the path of the heat flow is more important than the overall thickness of the tile.

Walls Nos. 63, 64, 66 and 91 reported in the paper are similar in that they are of 8-in. tile and have two large and one small air space. The coefficients of transmission are respectively 0.403, 0.299, 0.360 and 0.418 corrected for a wind velocity of 15 mph, thus indicating the variation that is likely to occur with similar types of tile, due apparently to surface finish, treatment of joints, workmanship, etc. The tile wall figures in Table 12, p. 44, THE GUIDE 1932, include 1-in. stucco on the exterior, but if computed without this finish, would have values of 0.33 for 8-in. tile with two air spaces in the direction of heat flow and 0.25 for 12-in. tile with three air spaces.

Wall No. 21 of the paper consists of ¾-in. stucco on 8-in. clay tile with an interior finish of ½-in. plaster on ½-in. insulating board on furring strips. This is similar to Wall No. 8A, Table 12, Chapter 3, THE GUIDE 1932, except that THE GUIDE construction includes 1 in. of stucco (which difference is, of course, negligible) and the form of the tile is different. The coefficient given in the paper is 0.18 corrected for a wind velocity of 15 mph, whereas THE GUIDE coefficient is 0.172 for the same wind velocity, but for two air cells in the path of the heat flow instead of two large and one small cell.

Wall No. 22 corresponds to Wall No. 2A in the same table except for the difference in the type of tile and the thickness of stucco. THE GUIDE value is 0.296 as compared with an average test value as given in the paper of 0.372, corrected to a wind velocity of 15 mph. Walls No. 24 (of the paper) and No. 4A (THE GUIDE) are also similar, the coefficient for the former being 0.233 (average of two tests and for a 15 mph wind velocity) and for the latter, 0.219. Except for the latter construction, the comparable values in THE GUIDE are lower.

There are no other tile constructions for which figures are given in THE GUIDE and in the paper which are sufficiently similar to be comparable.

Concrete Walls: It has been indicated in a previous paper[4] that the conductivity of 8.3, used for computing the heat transmission coefficients in THE GUIDE, is low and that the average value is probably about 12.0. Consequently, it is not surprising that the concrete values reported in the Rowley-Algren paper are consistently higher than in THE GUIDE. Curiously enough, the values for still air on both sides for 6 in. monolithic concrete check almost exactly the computed value in THE GUIDE for still air on one side and a wind velocity of 15 mph on the other. THE GUIDE value is 0.583, whereas the average of Walls 55, 68 and 69 is 0.585 for still air. When corrected for a wind velocity of 15 mph the latter coefficient becomes 0.786. As pointed out in this and the previous paper referred to.[1] the age of the concrete has an important bearing on its conductivity.

The conductances used in THE GUIDE for calculating the overall coefficients for walls constructed of cinder and concrete blocks were computed in the same manner as the hollow tile factors, using the proper air space values and conductivities of 8.3 and 5.2 per 1 in. for stone aggregate and cinder aggregate concrete, respectively. The 8-in. block values in THE GUIDE were based on one air cell in the direction of heat flow and the 12-in. values were based on two air cells. Since the 12-in. blocks for which tests are reported in the paper contained one air cell in the direction of heat flow, the factors for this material are not comparable. The comparative figures for the 8-in. blocks for a 15 mph wind velocity are as follows:

---

[4] See Conductivity of Concrete, by F. C. Houghten and Carl Gutberlet (A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932).

	The Guide 1932	Rowley-Algren
8-in. concrete blocks	0.458	0.557 (average 3 tests)
8-in. cinder blocks	0.311	0.422 (average 2 tests)

As pointed out in the paper, there is naturally a variation in cinders and any one test cannot be considered as a standard.

Brick Walls: The Rowley-Algren paper gives an overall coefficient of 0.499 for a single 3¾-in. thickness of common yellow brick for still air conditions. The factor for a single row of brick given in Table 25, Chapter 3, The Guide 1932 is 0.437 for still air on both sides. The Guide value in this case is 12.4 per cent lower than the test value reported in the paper.

The tests included two types of 8-in. brick walls, one of two courses of common yellow clay brick (Wall No. 51), for which a still-air value of 0.355 was obtained, and the other of one course of common yellow clay brick and one course of face brick (Wall No. 52), for which a still-air value of 0.406 was obtained. The overall coefficients for these two walls for a wind velocity of 15 mph were 0.423 and 0.496 (average of two tests), respectively. The value given in The Guide for this construction is 0.385, again somewhat lower than the test values for the same outside wind velocity and about midway between the still air and 15-mph wind velocity values for the two courses of common yellow brick.

Frame Wall: A frame wall (No. 37) consisting of face brick, a ¾-in. air space, building paper, wood sheathing, 1.1-in. thickness of flexible insulation, air space, and wood lath and plaster was included in the tests reported in the Rowley-Algren paper, and the coefficient obtained on four tests was 0.109 for a wind velocity of 15 mph. Although a factor is not given in The Guide for this identical construction, the coefficient for this construction, computed on The Guide basis, would be 0.112, which for practical purposes would be considered the same as the test value.

Gypsum Partition Tile Walls: Wall No. 81 consisting of 4-in hollow gypsum tile has a coefficient of 0.344 for still air on both sides. The value given in Table 25, p. 57, The Guide 1932, for a similar wall is 0.273 or 20.7 per cent lower. This discrepancy may be explained as follows: The conductance used for computing The Guide value was 0.46 for the 4-in. thickness, and this value was estimated from the air space data in Table 3, Chapter 3, The Guide 1932, assuming three cores per 12-in. width, each 2½ in. in diameter. The conductivity used for solid gypsum tile in arriving at the conductance of the 4-in. hollow gypsum tile was 1.66 per 1 in., which value is given in Table 7, Chapter 3, The Guide 1932, and is based on tests at the University of Minnesota. A conductivity of 2.96 for gypsum tile having a higher density is also given in this table. Had this value been used instead of 1.66, the computed and tested values would have more nearly checked. The surface coefficient used for computing The Guide coefficient was 1.34 for both sides (still air), whereas the surface coefficients given in the paper for the actual test were 1.671 and 1.542 for the inside and outside surfaces, respectively.

Rubble Walls: Values of 0.54 and 0.553 are given in the paper for plain 8-in. limestone walls for still air conditions (Walls Nos. 75 and 76), or 0.707 and 0.729, respectively when corrected for a wind velocity of 15 mph. The Guide value for this construction is 0.556 for a wind velocity of 15 mph which checks fairly well with the test values in the paper for still air, but is 21.4 per cent and 23.7 per cent lower than the two test values corrected for a wind velocity of 15 mph.

The conductivity for limestone used in The Guide was 10.00 as compared with the value of 12.5, which was the apparent value obtained in the paper assuming still air surface coefficients of 1.65.

Conclusions: The test values reported in this paper were in general higher than those found in The Guide for similar constructions. This was due not only to the fact that the conductivities and conductances used for computing The Guide values

were apparently lower than the values obtained in the tests reported in the paper, but the surface coefficients used in THE GUIDE were also lower than those obtained at the University of Minnesota.

An average still air surface coefficient of 1.34 was used in THE GUIDE for all surfaces, whereas the data reported in this and other papers resulting from the cooperative agreement between the Society and the University of Minnesota indicate that the average value for building materials is about 1.65. Likewise, the surface coefficient used in THE GUIDE of 4.02 for a wind velocity of 15 mph is lower than that used in the paper for correcting for this wind velocity.

Although the overall heat transmission coefficients in THE GUIDE are average lower than test values reported in the paper, the indications are that THE GUIDE values, when used in accordance with the customary procedure for estimating heat losses, result in more heating surface in most cases than is necessary. This, however, is probably due to the inadequate data for combining all of the factors entering into the calculations of the maximum heating load, that is, for ascertaining the worst combination of all of the many contributing elements of the problem. Not the least important of these is the heat capacity of heavy masonry walls, which results in a pronounced fly-wheel effect.

R. E. BACKSTROM (WRITTEN): In their paper, Professor Rowley and Mr. Algren have contributed a real service to science and to this Society in making available much valuable information on the over-all heat transmission coefficients of masonry walls. Their work should go far towards clearing up the age-old question of proper coefficients for walls constructed of masonry units containing relatively large air spaces.

One or two suggestions have occurred to me in reading through this paper. First, in connection with the frame walls, it is my reaction that the author's description of the materials used might be amplified. For example, the sheathing is described as being of "fir" and the siding of "pine." It is interesting to note that the Bureau of the Census, in its report, "The Principal Lumber Industries, 1929" lists five different species of fir under white fir, in addition to balsam and Douglas fir; ten species of pine under yellow pine, and four under white pine, in addition to western yellow pine. In view of the commercial importance of many different species of these woods, which doubtless vary in their resistance to heat passage, it would seem worth while to positively identify the material used in the tests.

I think it might be advisable also to construct the test specimens from lumber cut to American Lumber Standards. On this basis, 1-in. boards should measure 25/32 in. rather than 13/16 in. This may seem trivial, but there is a definite movement in America to standardize lumber dimensions, and I believe the Society should encourage that idea as much as possible.

The authors point to the difference in the coefficients of hollow tile walls having the same thickness but constructed of tile units of different design. That some of this difference is due to the height of the unit seems probable, and I do not believe this fact has been brought out. For example, comparing Walls 59 and 64, we find that both are built of 8 in. tile, similar in design, especially as regards the type of horizontal joint, but the units in Wall 59 are 5 in. "high" whereas those in Walls 64 are 10¼ in. Obviously, twice as many horizontal mortar joints occur in the wall composed of the small units, and we would naturally expect that wall to have the higher heat transmission factor. Upon examining the coefficients, we find this to be the case. Quoting from the paper, the values are 0.280 and 0.265, respectively.

My only other suggestion has to do with a more complete description of the masonry units used in these tests. In his papers entitled The Water Absorption and Penetrability of Brick, *Proceedings A. S. T. M.* Vol. 29, Part II, 1929, and The Compressive and Transverse Strength of Brick, Bureau of Standards *Journal of*

Research, Vol. 2, April 1929, J. W. McBurney, formerly Research Associate, *Common Brick Manufacturers Association of America*, gives important data on the properties of a variety of brick representing different methods of manufacture. He points out that brick are made from either clay or shale; that the methods of manufacture are known as dry press, semi-dry press, soft mud, or stiff mud; and that brick are graded (according to the rate of water absorption) as vitrified, hard, medium, or soft. In view of the great variety of brick, a knowledge of important facts in connection with those used in the authors' tests, and also such other properties as may be expected to influence thermal conductivity, notably specific gravity and porosity, would be helpful to engineers and architects using these data, and to other investigators in correlating the results of various tests. No doubt much of this information is already in the authors' possession and will be reported in future papers on the subject.

JOHN S. BUSCH: This is an interesting paper and should be of much value to the building industry. We have lately been able to make similar tests on wall sections. It is our opinion that actual wall section test values are much to be preferred to those calculated from hot plate tests. There are many extraneous factors which are brought into play under actual conditions that are not taken into consideration in calculations made from hot plate tests.

The tests on an 8-in. brick wall, an 8-in. rubble wall and many of the other walls check quite well with the hot plate tests. On the other hand, the wall tests on a 1-2-4 concrete mix run from 11 to 12 as compared to 8.30 given in THE GUIDE 1932. In our work we have often run across similar discrepancies, particularly with the fill types of insulation or in general with any type of insulation material which is very porous and not sealed to prevent circulation of air currents.

The results obtained on Tests 21, 37 and 42 are very illuminating since they very clearly show the necessity of a certain amount of a highly efficient insulation material in all types of wall construction. These three walls are the only ones which approach the desired degree of heat loss resistance.

The resistance shown for air spaces enclosed in paper, aluminum bronze and aluminum foil surfaces are very interesting. We have done some work along this line and these tests check our results quite closely.

Fig. 10 which shows the change in thermal conductivity for changes in mean temperature for various types of wall construction is very useful. It will be noted that the decrease in thermal conductance (U) per one degree drop in mean temperature of these walls runs as follows:

No. 55—6-in. solid concrete	0.15%
No. 57—Lath and plaster partition	0.22%
No. 8—Uninsulated frame wall	0.33%
No. 12—Frame wall with insulation *C* applied between studs	0.46%

For some time it has been the custom to use a correction factor of 0.15 per cent for a 1 F change in mean temperature. We have often found in our work that this factor is too small to fit the facts. Professor Rowley's Fig. 10 substantiates our findings. It will be noticed particularly that the slope becomes steeper and the factor becomes larger as the number of air spaces or the roughness of the surface is increased. Maximum advantage of this greater increase in efficiency is obtained only when the insulation is applied between studs in a manner such that the air space is divided.

PRESIDENT ROWLEY: A point of particular interest is the effect of air spaces in tile wall construction. It has been difficult in the past to determine just what values should be given to such air spaces. It is evident that a rather large percentage of the heat flows directly through the joints or solid material of the tile wall. The results show that those walls which have a staggered air space construction, and

thus a longer path for the heat flow through the solid portion of the wall, give better results than those walls which have a direct path for the heat flow.

MR. ALGREN: Referring to Mr. Backstrom's discussion, I would like to say that in the walls tested white pine was used for the siding and Douglas fir for the sheathing. The measurements were taken of the material as applied. The face brick would be classified as hard brick and common clay brick as soft brick.

No. 938

TESTS OF CONVECTORS IN A WARM WALL TESTING BOOTH

By A. P. Kratz [1] (*MEMBER*), M. K. Fahnestock [2] (*NON-MEMBER*) and E. L. Broderick [3] (*NON-MEMBE* , Urbana, Ill.

This paper is the result of research conducted at the University of Illinois in cooperation with the A. S. H. V. E. Laboratory

The data presented in this paper were obtained in connection with an investigation conducted by the Engineering Experiment Station of the University of Illinois, of which Dean M. S. Ketchum is the Director. This work is carried on in the Department of Mechanical Engineering under the direction of A. C. Willard, Professor of Heating and Ventilation and head of the department. This paper includes part of the results from one year's work constituting a continuation of a general research program devoted to the study of heating rooms with various types of direct steam and hot water radiators and convectors and the material will be incorporated in a bulletin of the Engineering Experiment Station.

THE A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code)[4] presented at the Semi-Annual Meeting of the Society, Swampscott, Mass., June, 1931, provides for tests to be run in a warm wall booth of stated construction and dimensions. It further specifies that the standard temperature for the steam in the heating unit shall be 215 F and for the air at the inlet, 65 F, and that the temperature of the air at the inlet shall be not less than 60 F nor more than 80 F. For tests run with inlet air temperatures other than 65 F, but conforming with these limiting temperatures, the heat output obtained by test shall, according to this code, be reduced to a standard heat output by multiplying the actual heat output by the correction factor:

$$C = \left[\frac{150}{t_s - t_1}\right]^{1.3} \qquad (1)$$

[1] Research Professor in Mechanical Engineering, University of Illinois.
[2] Special Research Assistant Professor in Mechanical Engineering, University of Illinois.
[3] Research Assistant in Mechanical Engineering, University of Illinois.
[4] A. S. H. V. E. Transactions, Vol. 37, 1931, p. 367.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Milwaukee, Wis., June, 1932, by M. K. Fahnestock.

where

150 = the temperature difference between steam at 215 F and inlet air at 65 F

t_s = the temperature of steam during the test

t_i = the inlet air temperature during the test

This correction factor is the one commonly used in the case of cast-iron radiators and its validity has been well established when used in this connection.

At the time the A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code) was presented, two points remained obscure to a certain extent. The validity of extending the application of the correction factor derived for cast-iron radiators to include convectors had not been definitely proved. A correlation between convector tests run in a warm wall booth and similar tests run in rooms simulating actual service conditions had not been experimentally demonstrated. The testing program outlined in this paper was therefore undertaken in order to obtain information concerning these two points. For this purpose 225 tests were run, including tests on 4 distinct types of convectors and one type of cast-iron radiator, with each convector or radiator tested both in the warm wall booth and under service conditions in the low temperature testing plant, or more briefly designated as the *cold room*.

DESCRIPTION OF APPARATUS

An elevation section of the low temperature testing plant is shown in Fig. 1. This plant has been fully described in previous publications.[5] The test rooms in this plant were 9 ft by 11 ft with 9-ft ceilings.

A vertical section of the warm wall test booth showing the location of the booth with respect to the large room of the Mechanical Engineering Laboratory, in which it was erected, is shown in Fig. 2, and a detail of the piping for the heating unit and the apparatus for measuring steam condensation is shown in Fig. 3. This booth was constructed in accordance with the specifications given in the A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code) and was 12 ft by 13 ft 4 in., with a 9-ft ceiling. For a few tests, the 2-pipe steam connection shown for the warm wall booth in Fig. 3 was replaced with a 1-pipe connection similar to the one shown for the low temperature testing plant in Fig. 1.

In both the test room and the test booth, the room temperature at various levels was measured on the central vertical axis of the room or booth. The temperature of the air at the inlet of the convector was measured at a distance of 3 in. in front of the inlet at at least three points spaced from 8 to 12 in. apart, midway between the top and bottom of the inlet opening. In the case of the test room in the low temperature testing plant all temperatures were measured by means of thermocouples in order to avoid any necessity for entering the rooms and thus disturbing the test conditions. In the case of

[5] University of Illinois, *Engineering Experiment Station Bulletins* Nos. 192 and 223; Investigation of Heating Rooms with Direct Steam Radiators Equipped with Enclosures and Shields, by A. C. Willard, A. P. Kratz, M. K. Fahnestock and S. Konzo (A. S. H. V. E. TRANSACTIONS, Vol. 35, 1929); Steam Condensation an Inverse Index of Heating Effect, by A. P. Kratz and M. K. Fahnestock (A. S. H. V. E. TRANSACTIONS, Vol. 37, 1931); Performance of Convector Heaters, by A. P. Kratz and M. K. Fahnestock (A. S. H. V. E. TRANSACTIONS, Vol. 38, 1932).

the test booth, all temperatures were measured by means of mercury thermometers as specified in the code. In this case, the booth had one open side, and was surrounded by temperature conditions similar to those in the booth itself. Hence, no disturbance was caused by the observer entering the booth.

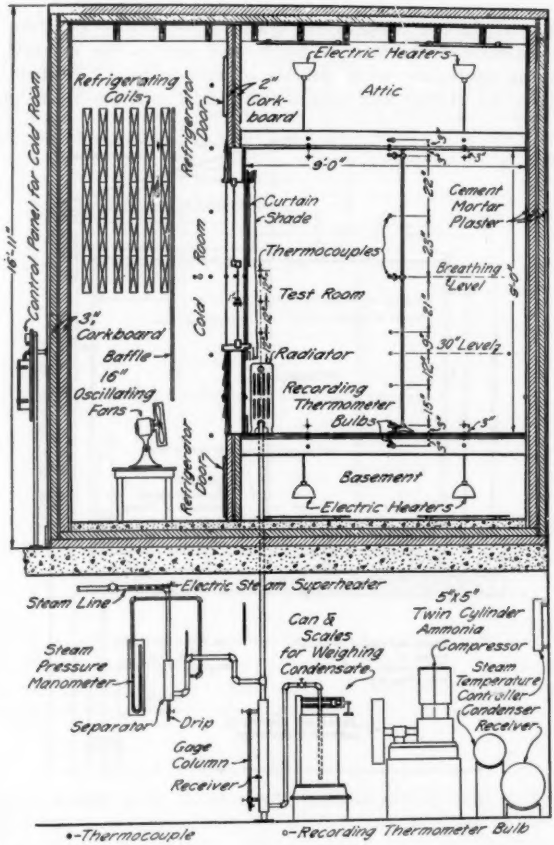


FIG. 1. ELEVATION SECTION OF LOW TEMPERATURE TESTING PLANT

Four distinct types of convectors and one type of cast-iron steam radiator were tested. For 3 of the types of convectors, 2 different sizes of heating units were tested for each type, thus making in all 8 different units tested. The types and dimensions of all convectors are shown in the table in Fig. 4 and in the insets in Figs. 5 to 12.

Test Procedure

Each convector was tested in both the low temperature testing plant and the warm wall test booth with the full amount of heating surface effective; that is, no part of the heating surface or air passages was either wrapped or blocked.

In the case of the low temperature testing plant, the temperature in the cold room was maintained at about —2.0 F and one of the exposed walls was subjected to an equivalent wind velocity of approximately 10 mph. The temperature above the ceiling was maintained at 62 F and the air in the basement

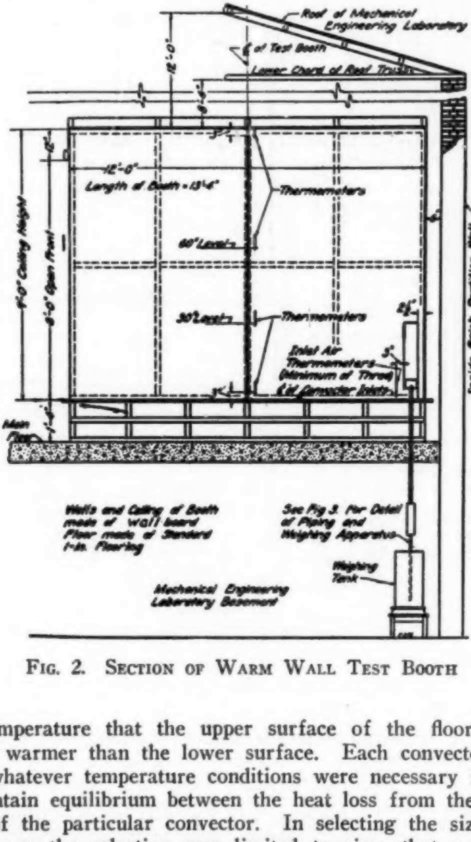


Fig. 2. Section of Warm Wall Test Booth

at such a temperature that the upper surface of the floor was approximately 2 deg warmer than the lower surface. Each convector was allowed to establish whatever temperature conditions were necessary in the room in order to maintain equilibrium between the heat loss from the room and the heat output of the particular convector. In selecting the sizes of the convectors, however, the selection was limited to sizes that would not either overheat or underheat the room an unreasonable amount; that is, temperatures above 75 F or below 60 F at the 30-in. level were considered as highly undesirable.

In the case of the test booth it was considered advisable to obtain a curve for each convector, establishing the relation between the steam condensation and the temperature of the air at the inlet of the convector. Tests were therefore run at different inlet air temperatures, varying over a range of 60 F to 90 F. In order to accomplish this, the large room in which the test booth was erected was heated or cooled to a temperature approximating the desired inlet temperature, and the convector was allowed to establish the temperature conditions in the booth necessary for thermal equilibrium.

For both the test room and the test booth, no test observations were made until conditions had remained constant for several hours, as indicated by readings of all thermocouples or thermometers. When the required thermal constancy had been attained, the condensate was weighed over a period of 1 hr,

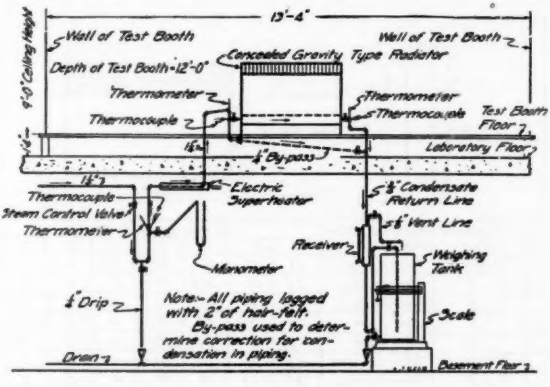


FIG. 3. DETAIL OF PIPING FOR WARM WALL TEST BOOTH

and no test was accepted if the condensate showed more than 2½ per cent deviation in the successive 10-min increments of weight. At the end of each test, a separate test was run to determine the condensation in the piping alone, and the total condensate was corrected by subtracting the amount so determined.

RESULTS OF TESTS

The results of the tests made in the warm wall booth for all of the heating units tested are shown as full line curves in Figs. 5 to 12. These tests are represented by the unnumbered points, and the curves indicate the mean trend of these points. Additional points shown near each curve, specifically designated by test numbers, represent the results from the tests made in the room of the low temperature testing plant. Duplicate tests were always made in the latter case, and when no change in conditions occurred, greater differences than 2.0 per cent in the steam condensations for the two tests were never obtained. The differences in temperature between the steam in the heating unit and the air at the inlet of the enclosure have been used as the abscissae for all curves, but since the tests were all run with a steam temperature of 216.5 F these abscissae are also representative of the temperature of

the air at the inlet, and the latter may be obtained by subtracting any temperature shown from 216.5 F.

In the case of the 8-section, 26-in., 5-tube unenclosed cast-iron radiator shown in Fig. 5, the heat output has been plotted against the difference between the temperature of the steam and the air 3 in. above the floor at the center of the booth or room. Owing to the disturbing influence of the radiator itself, temperature measurements made in the immediate vicinity of the radiator were of uncertain value, and the temperature of the air 3 in. above the floor at the center of the room was comparable with the temperature of the air at inlet observed for the convectors. In comparing Fig. 5 with Figs. 6 to 12, it is evident that the unenclosed radiator was more sensitive to stray

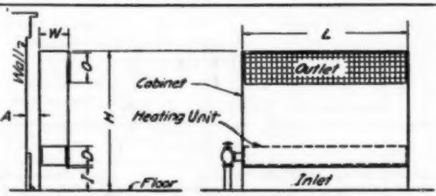


Fig. No	Convector No.	Type of Heating Unit	Dimensions in Inches						
			A	W	H	L	I	O	D
5	8-Sec, 26", 5 Tube Radiator		2½	8	26	20	—	—	—
6	2	Non-ferrous	2½	5	30⅛	31	6	6	6
7	3	Non-ferrous	2½	7½	30½	21	6	6	6
8	1	Cast Iron	2½	5½	26	37½	4	5	14⅜
9	13	Cast Iron	2½	4	26	37½	3	5	14⅜
10	6	Non-ferrous	2½	5½	28	30	4	5	2
11	5	Non-ferrous	2½	5½	18	36	4	5	2
12	7	Bimetallic	2½	6 1/16	28½	22⅜	4¼	3¼	20

FIG. 4. DIMENSIONS OF HEATING UNITS TESTED

air currents than the convectors, but it was possible to draw a curve representing a fair average trend for the points.

In order to determine the probable deviation of the corrected heat outputs, as provided for in the A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code), from the values that would have been obtained by tests at various temperature ranges between steam and inlet air, the calculated curves, shown as broken lines, have been superimposed on the test curves in Figs. 5 to 12. Since the standard temperature range was defined as from 215 F to 65 F, or 150 deg, both the actual and the calculated curves would be coincident at this point. Hence, by passing a curve defined by the equation

$$ {}_{c}H = {}_{a}H_{150} \left[\frac{t_s - t_1}{150} \right]^{1.3} \qquad (2) $$

through the point on the test curve corresponding to the temperature difference of 150 deg, the difference between the curves at any given temperature differ-

ence would represent the approximate error in the application of the correction factor to test results obtained with that temperature difference. In Equation 2,

$_cH$ = the calculated heat output at any temperature difference between steam and inlet air, Btu per hour

$_aH_{150}$ = the actual heat output read from the test curve at a temperature difference of 150 deg

t_s = the steam temperature for which $_cH$ is to be calculated approx. 215.0 F

t_i = the inlet air temperature for which $_cH$ is to be calculated

The deviation, or per cent error, in the application of the correction factor at the two limiting values for the temperature of the inlet air, 60 F and 80 F, as specified in the A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code), is given for each heating unit in Table 1. From the table it may be noted that with the exception of Convector No. 5, the correction factor is applicable to all of the heating units

TABLE 1. DEVIATION OF CALCULATED CORRECTION FROM TEST CURVE

Fig. No.	Heating Unit No.	Type of Heating Unit	60 F Inlet				80 F Inlet			
			Heat Output Btu per Hour		100 x Ratio $\frac{_cH_{155}}{_aH_{155}}$	Per Cent Diff.	Heat Output Btu per Hour		100 x Ratio $\frac{_cH_{135}}{_aH_{135}}$	Per Cent Diff.
			By Test $_aH_{155}$	Calculated $_cH_{155}$			By Test $_aH_{135}$	Calculated $_cH_{135}$		
5	..	5-tube C.-I. Radiator	6745	6770	100.3	+0.3	5715	5665	99.1	−0.9
6	2	Non-ferrous	5830	5810	99.7	−0.3	4820	4860	100.8	+0.8
7	3	Non-ferrous	5200	5185	99.7	−0.3	4310	4340	100.7	+0.7
8	1	Cast Iron	6270	6275	100.1	+0.1	5260	5245	99.7	−0.3
9	13	Cast Iron	5680	5670	99.8	−0.2	4700	4740	100.8	+0.8
10	6	Non-ferrous	6130	6115	99.8	−0.2	5060	5115	100.9	+0.9
11	5[a]	Non-ferrous	5575	5515	98.9	−1.1	4410	4605	104.4	+4.4
12	7	Bimetallic	6780	6805	100.4	+0.4	5740	5695	99.2	−0.8

[a] Overall height = 18 in.; overall height of all other units 26 in. to 30⅜ in.

with an accuracy within 1 per cent over the specified range of inlet air temperatures of from 60 F to 80 F. In most cases, particularly in that of the unenclosed 5-tube cast-iron radiator, a slight shift in the slope of the test curve would have brought the two curves into coincidence over the whole range, and the new test curve would still be a fair representation of the trend of the points. No such shift was made, however, but the test curves were allowed to remain as first drawn, uninfluenced by the trend of the calculated curves.

The maximum deviation of 4.4 per cent occurred with Convector No. 5 which was of the same type as No. 6, for which the maximum deviation was only 0.9 per cent, and hence the difference could not be attributed to type. However, this convector had the lowest overall height of any of the heating units tested. It was only 18 in. high, while the other heating units, including

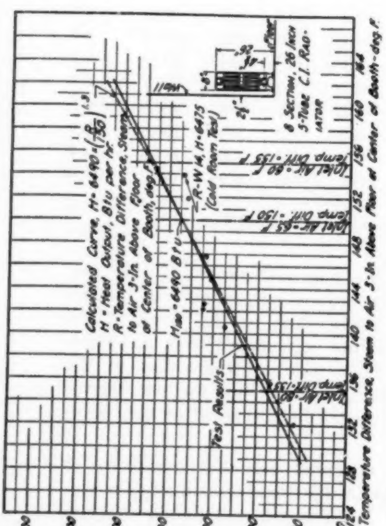


FIG. 5. PERFORMANCE CURVE FOR 8-SECTION, 26-INCH, 5-TUBE C. I. DIRECT RADIATOR

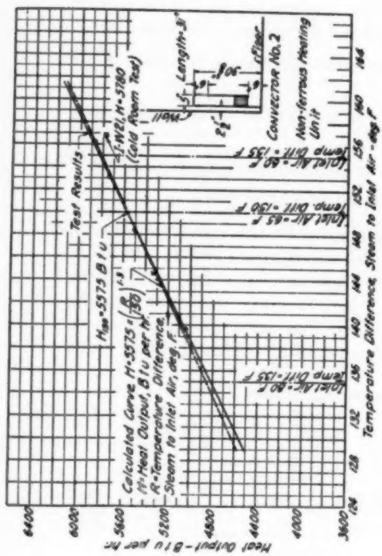


FIG. 6. PERFORMANCE CURVE FOR CONVECTOR NO. 2

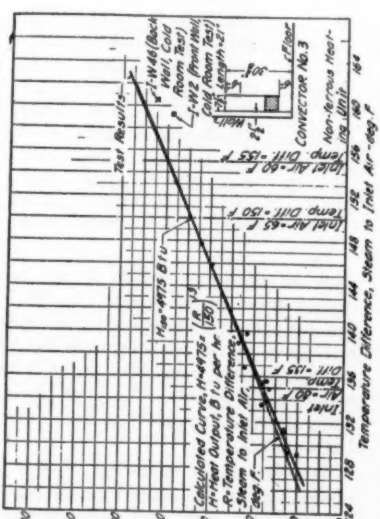


FIG. 7. PERFORMANCE CURVE FOR CONVECTOR NO. 3

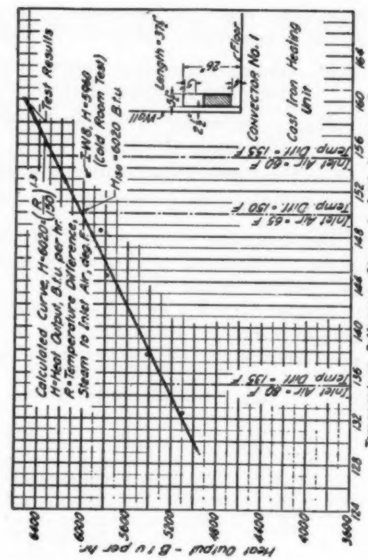


FIG. 8. PERFORMANCE CURVE FOR CONVECTOR NO. 1

the unenclosed cast-iron radiator, varied from 26 in. to 30⅜ in. Hence, it is possible that the use of the correction factor may not be applicable to convectors having overall heights outside of the range of from 25 in. to 32 in., and that further work is advisable using overall heights outside of this range.

For the discussion of the correction factor, Equation 2 has been applied to the test results obtained with a temperature difference of 150 F between steam and inlet air in order to compute the heat output that would have been obtained with a temperature difference of 135 deg, corresponding to an 80-F inlet air temperature, assuming the correction factor to be valid. In the actual application of the correction factor to test results the reverse process would be used. That is, the equation would be applied to the actual test results obtained with a temperature difference of 135 deg in order to compute an hypothetical heat output that would have been obtained with the standard tem-

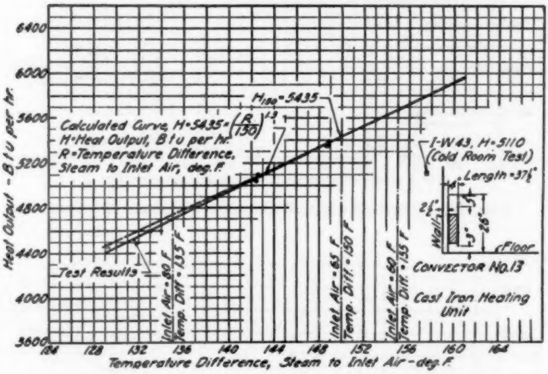


FIG. 9. PERFORMANCE CURVE FOR CONVECTOR NO. 13

perature difference of 150 deg. Since the base value in Equation 2, ${}_aH_{150}$ in the one case and ${}_aH_{135}$ in the other case, would be different, the two calculated curves would not be exactly parallel, and the difference between the actual and calculated curves at a temperature difference of 135 deg in the first case would not be identical with the difference between the actual and calculated curves at a temperature difference of 150 deg in the second case. This is illustrated in Fig. 13, from which it is evident that the difference in the two methods of treatment is purely academic since one method gives a deviation of 4.4 per cent while the other gives 4.2 per cent.

A comparison of the heat outputs of the various units when tested first in the low temperature test plant and then tested in the warm wall booth with the same temperature for the air at inlet, may be obtained by comparing the positions of the numbered points with the positions of the curves for the booth tests in Figs. 5 to 12. For convenience, these results have been consolidated in Table 2, the last column of which shows the ratio of the heat output obtained in the low temperature testing plant to the heat output in the warm wall booth for the same temperature of air at the inlet. No very definite

correlation seems to exist, but it is evident that the warm wall booth tests give heat outputs ranging from 2.4 to 12.0 per cent higher than those obtained in the low temperature testing plant under actual service conditions.

Practically all of the tests in the warm wall booth were run with the 2-pipe steam connection shown in Fig. 3, while in the low temperature testing plant a

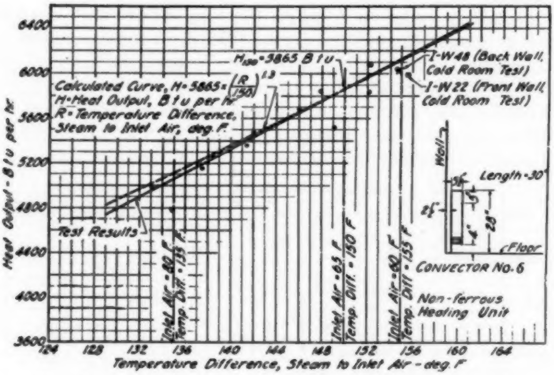


FIG. 10. PERFORMANCE CURVE FOR CONVECTOR NO. 6

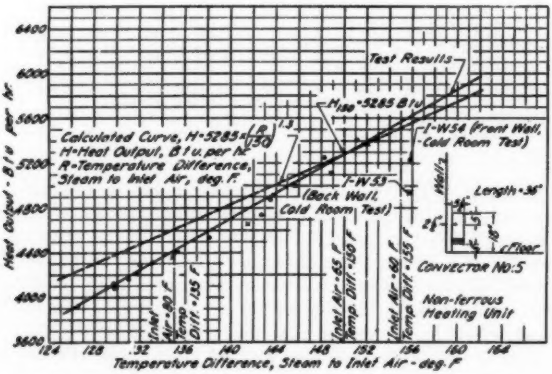


FIG. 11. PERFORMANCE CURVE FOR CONVECTOR NO. 5

1-pipe connection was used. In order to determine whether the difference in heat outputs could be attributed to this difference in piping connections, the 2-pipe connection in the warm wall booth was replaced with a 1-pipe connection identical with the one shown in Fig. 1, and a series of tests was run on each of Convectors Nos. 2, 5 and 7. These results are shown by the crosses in Figs. 6, 11 and 12. The location of these points with respect to

the other points indicates that no difference in results could be attributed to the difference in steam connections.

During the test periods in the low temperature testing plant the rooms were closed and readings were made with thermocouples wired to a potentiometer located outside of the rooms. Hence, no disturbance in conditions could be

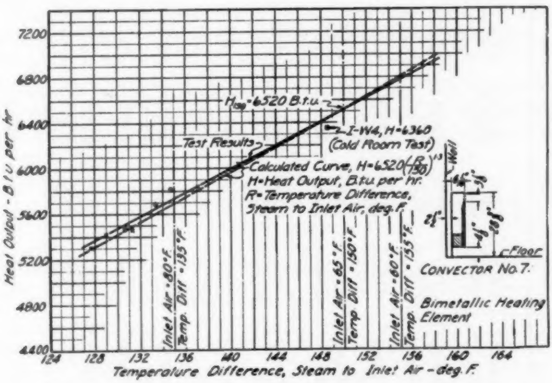


Fig. 12. Performance Curve for Convector No. 7

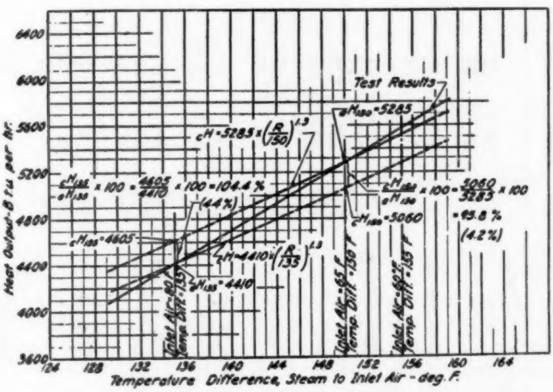


Fig. 13. Sample Application of Correction Factor

brought about by observers entering the rooms, or by air currents in the main laboratory. In the case of the warm wall booth, the observer entered the booth to read thermometers, and since one side of the booth was open, it was possible that air currents in the main laboratory might be reflected in the booth. A desk fan was therefore placed in the booth and located in such a way that, while the air current was not directed against the convector, the disturbance

TABLE 2. COMPARISON OF PERFORMANCE OF CONVECTORS TESTED IN WARM WALL BOOTH AND IN LOW TEMPERATURE TESTING PLANT

Fig. No.	Test No. (Cold Room)	Con-vector No.	Type of Heating Unit	Overall Dimensions, Inches			Location	Btu Output per Hour	Inlet Air Temp., (Deg. Fahr)	Temp., Diff. Steam to Air	30-in. Level Temp., (Deg Fahr)	60-in. Level Temp., (Deg Fahr)	$\frac{H_{\text{Cold Room}}}{H_{\text{Booth}}} \times 100$
				Width	Height	Length							
5	*R*-W14		8-Sec. 26", 5-Tube, C. I. Rad.	8	26	20	Cold Room [c]	6475	62.6[a]	153.9[b]	71.3	78.5	96.8
			" "	8	26	20	Booth	6690	62.6[a]	153.9[b]	....	....	....
6	*I*-W21	2	Non-ferrous	5	30⅝	31	Cold Room [c]	5780	59.7	156.8	67.3	76.4	97.6
		2	"	5	30⅝	31	Booth	5920	59.7	156.8	....	....	....
7	*I*-W2	3	Non-ferrous	7½	30⅝	21	Cold Room [c]	5150	57.3	159.2	63.5	73.6	95.6
		3	"	7½	30⅝	21	Booth	5385	57.3	159.2	....	....	....
	I-W46	3	"	7½	30⅝	21	Cold Room [d]	5300	56.0	160.5	62.7	73.4	97.3
		3	"	7½	30⅝	21	Booth	5445	56.0	160.5	....	....	....
8	*I*-W8	1	Cast Iron	5½	26	37½	Cold Room [c]	5960	63.2	153.3	70.8	77.4	96.4
		1	"	5½	26	37½	Booth	6185	63.2	153.3	....	....	....
9	*I*-W43	13	Cast Iron	4	26	37½	Cold Room [c]	5110	58.9	157.6	65.1	72.2	88.0
		13	"	4	26	37½	Booth	5810	58.9	157.6	....	....	....
10	*I*-W22	6	Non-ferrous	5½	28	30	Cold Room [c]	5990	60.9	155.6	69.2	77.8	97.2
		6	"	5½	28	30	Booth	6165	60.9	155.6	....	....	....
	I-W48	6	"	5½	28	30	Cold Room [d]	6025	61.7	154.8	70.1	79.5	97.8
11	*I*-W54	5	Non-ferrous	5½	18	36	Cold Room [c]	5230	60.7	155.8	67.8	73.5	93.1
		5	"	5½	18	36	Booth	5620	60.7	155.8	....	....	....
	I-W53	5	"	5½	18	36	Cold Room [d]	4940	60.7	155.8	67.8	74.8	87.9
12	*I*-W4	7	Bimetallic	6 1/16	28⅝	22⅝	Cold Room [c]	6360	67.9	148.6	71.2	80.1	99.0
		7	"	6 1/16	28⅝	22⅝	Booth	6440	67.9	148.6	....	....	....

[a] Temperature of air 3 in. above floor at center of room or booth.
[b] Temperature difference between steam and air 3 in. above floor at center of room or booth.
[c] Front wall.
[d] Back wall.

of the air in the booth was much greater than any that could be brought about by the observer entering the booth, or by stray air currents occurring in the large room in which the booth was located. The results from 2 tests indicated that the booth was comparatively free from the influence of stray air currents. Furthermore, no evidence could be obtained to indicate that the air motion in the large room was tending to circulate air into the booth at the bottom and out at the top, thus acting as a pump to increase the air circulation through the radiator. Therefore, differences in test results can hardly be attributed to these causes.

The essential difference between the influence of the rooms in the low temperature testing plant and of the warm wall booth on the performance of the heating units seems to arise from the fact that in the test room the heating

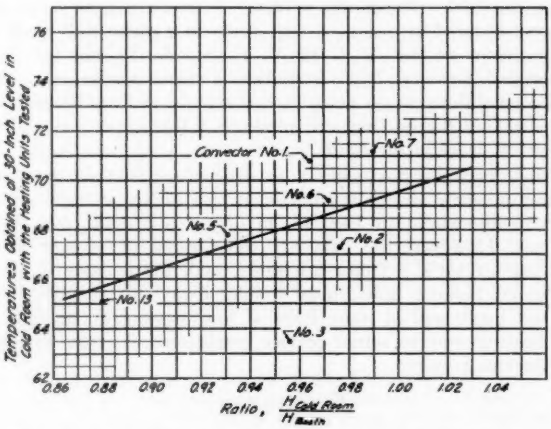


FIG. 14. CORRELATION CURVE FOR PERFORMANCE IN LOW TEMPERATURE TESTING PLANT AND WARM WALL BOOTH

unit must establish its own environment, limited by the necessity for establishing thermal equilibrium between the heat output from the heating unit and the heat loss from the room, while in the warm wall booth the heat generated can escape freely, and the environment is established more or less independent of the heat output of the heating unit. This is reflected to a certain extent by the fact that the temperature gradient from floor to ceiling in the room in the low temperature testing plant was much greater than that in the warm wall booth. Just what bearing this has on the performance of the heating units when the temperatures of the air at the inlets are the same in the two cases, is not apparent, but it seems probable that the heating units tested in the low temperature testing plant must all be of such a size as to produce some definite temperature at a definite level in the test room, in order for the heat output to be comparable with that obtained in the warm wall booth.

In order to determine whether such a correlation was possible, the ratio of the heat outputs obtained in the low temperature testing plant to those obtained

in the warm wall booth were plotted against the corresponding temperatures at the 30-in. level obtained in the low temperature testing plant, as shown in Fig. 14. While the correlation can not be considered as very satisfactory, some marked tendencies are indicated. As the temperature at the 30-in. level in the test room became higher, the ratio of the heat outputs for tests run with the same inlet air temperature in the test room and the booth also became higher, and this ratio approached 1.0 for a temperature of approximately 69 F at the 30-in. level. The ratio for Convector No. 3 seems out of line with the others, but even taking this into consideration, it seems evident that if all of the heating units had been of such a size as to produce a temperature of 68 or 69 F at the 30-in. level in the test room of the low temperature testing plant, the heat outputs obtained in this plant would have all been within 10 per cent of those obtained in the warm wall booth with the same temperature for the air at the inlet. Hence, the evidence from these tests seems to indicate that, if a heater is rated by means of tests made in a warm wall booth according to the specifications of the A. S. H. V. E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code), the heat output under service conditions in an actual room will be within 10 per cent of the rating, provided that the heating unit produces a temperature of approximately 68 F at the 30-in. level in the room.

It is possible that the heat output as determined in the warm wall booth should be multiplied by an application or service factor of approximately 0.95, as indicated by the curve in Fig. 14, in order to provide for deviations in actual service. However, the data are too limited at present to make the recommendation of such a factor advisable, and, particularly in the cases of unusual types of heating units, tests should always be required both in the warm wall test booth and in some form of low temperature testing plant subjecting the heating unit to actual service conditions, in order to establish application or service factors.

CONCLUSIONS

The following conclusions may be drawn as applying to the data obtained in these tests:

1. The correction factor, $C = \left[\frac{150}{t_s - t_1}\right]^{1.3}$ for reducing the heat output of a convector under test conditions to the heat output under standard conditions, with steam temperature of 215 F and inlet air temperature of 65 F, is applicable within a probable error of 1 per cent for temperatures of the air at the inlet between 60 F and 80 F, and for overall heights of from 25 in. to 32 in. with a steam temperature of approximately 215 F.

2. For overall heights outside of the range of 25 to 32 in. it is possible that the error resulting from the application of the correction factor does not exceed 5.0 per cent.

3. The heat output of a convector under service conditions in an actual room will be within 10 per cent of the heat output determined from tests in a warm wall booth, with the same temperature for the steam and for the air at the inlet in both cases, provided that the size of the convector is sufficient to heat the actual room to a temperature of approximately 68 F at the 30-in. level.

DISCUSSION

R. M. CONNER[6]: The research work reported by Professor Kratz appears to be of the same excellent character as previous work of a similar nature carried out under the direction of Professor Williard. The *cold* room referred to is evidently the same one used in earlier work on radiators and radiator shields.

As stated, the three conclusions drawn are well established by the test data. The statements, however, are somewhat limited. The first conclusion establishes the correction factor, $C=\left(\frac{150}{t_s-t_1}\right)^{1.3}$, for a standard steam temperature, set limits of inlet air, and for a certain range of overall heights. It is probably not important to establish the factor for varying steam temperatures, since that condition can be readily maintained. The factor would still be expected to hold, nevertheless.

The second conclusion merely asserts that for overall heights outside of the range of 25 to 32 in., the deviation from the correction factor may be anything. Test data in this respect were taken on only one convector and the results are rather difficult to understand. Further research could very well be directed for the purpose of securing further data on this point.

The third conclusion relates test conditions to service conditions. Ten per cent is the tolerance stated even for specified conditions and appears rather broad. From the data, there appears to be no definite relationship. The points on Fig. 14 are well dispersed.

The *American Gas Association* Laboratory has had no experience with specific appliances mentioned and, hence, has never had quite the same problems. It has, however, made use of both a *cold* room and a *warm wall* booth in the testing of gas steam radiators. Convectors would not seem to present as difficult a problem as gas steam radiators, due to the double nature of the latter and the presence of radiation effects.

In the low temperature or *cold wall* room, the air temperature was kept at 70 F at the 60 in. level and the steam pressure in the radiator at 10 lb gauge. Equilibrium was produced by refrigerating the outside of the room, and the condensing power of the radiator measured. After equilibrium had been well established, readings were taken over a 2 hour period. This practice followed recommendations in the A.S.H.V.E. Code. It was felt that such conditions formed a standard basis and gave results which could easily be duplicated.

When tests were later made in ordinary Laboratory rooms and in a specially constructed booth, absolutely no direct correlation could be obtained. Deviations varied widely according to the nature of the radiator and also for the same radiator under different conditions. To surmount this difficulty, the condensation rates obtained in the *cold* room were regarded as standard and other radiators compared with these standard radiators in simultaneous tests in the booth. Very satisfactory results were obtained in this manner.

Regarding the correction factor, $C=\left(\frac{150}{t_s-t_1}\right)^{1.3}$, a similar factor was found to apply for steam condensation tests of gas steam radiators and also to apply for gas fired operation under narrow room temperature limits.

H. F. HUTZEL (WRITTEN): The results of this investigation are predicated upon the assumption that the Code as presented at the Semi-Annual Meeting in Swampscott, will assure a true reflection of a convectors performance. It is my impression

[6] Director, *American Gas Assn.* Testing Laboratory.

that this Code permits of a practice which will result in inaccuracies and that these inaccuracies are in evidence in tests conducted by the authors.

It has been my experience that with mercury thermometers located as prescribed by the Code for measuring inlet air temperatures, that shielded thermometers indicate temperatures of from 1½ to 3 deg lower than unshielded thermometers. Since the correction factor formula as shown in this report is effected inversely by the difference between the steam and inlet air temperatures, convectors tested with shielded thermometers will show a lower corrected capacity than those tested with unshielded thermometers.

Assuming this statement to be correct, and assuming for example that the true inlet air temperatures for convector 2 to be 2 deg lower than indicated, then the true performance curve would be parallel to and lie 2 deg horizontally to the right of curve as shown in Fig. 6. It is interesting to note that a curve so drawn very nearly passes through the point representing the cold room test. (I-W-21, H=5780)

By applying this correction to the hot booth test shown in Table 2 opposite Fig. 6, the corrected capacity would be 5820 Btu's instead of 5920, or only six tenths of 1 per cent higher than the corresponding cold room test.

With reference to the discrepancy between actual and calculated capacity curves, this cannot be attributed to the possible error indicated. Changing the base value alone would not change the slope of the calculated rating curve. It is my impression that the difference in the slopes of these two curves is an indication that the correction factor formula as shown does not apply to convector heaters even though its validity has been well established when used in connection with direct radiation.

T_1 in the formula for direct radiation, has reference to the room temperature at the breathing line whereas T_1 in the formula for convector heaters has reference to the temperature taken near the floor at the entrance to the heater. Furthermore the air coming in contact with a direct radiator, approaches it horizontally throughout its height and at various temperatures in accordance with temperature stratification. In the case of the convector heater the air enters same near the floor line only at the temperature indicated by the T_1. In view of this is it not inconceivable that one and the same correction factor formula should apply to both? It would seem more logical that the exponent in this formula should be lower than 1.3 deg and possibly unit since the conditions of operation for a convector heater are somewhat similar to a unit heater for which the correction factor formula has an exponent of unity.

When, however, the correction factor formula with unity is applied to the tests as reported in this paper, the calculated capacity curves are at greater variance than the calculated capacity curve shown. This accordingly does not substantiate my theory.

In view of the fact that this Society is giving consideration to the acceptance of a Standard Code for Testing Concealed Heaters, this is a very important paper and seems to indicate that further investigation should be made as to the validity of the present correction factor formula to convector heaters.

WARREN EWALD: I have a criticism which is made offhand and therefore to be valued accordingly. The difference in these two methods of testing the radiators was largely ascribed to the current of cold air in the cold room test flowing counter to the flow of air from the heater. It is possible that this has an actual effect in that direction, but certain tests I have made indicate that there is a greater effect from the loss of air flow through the convection heater.

In the tests described the convector casing is against a cold wall which is the same as having one cold wall on a chimney or stack. By reducing the temperature of the air in the stack the specific weight of this air is being lowered and naturally a smaller amount of air is caused to flow. I believe that this effect is greater than the effect of these counter currents. Also I think that this effect is proportional

to the heat losses through the wall of this cold room and that these experiments perhaps give a measure of the heat loss through this wall rather than an output of the convection heaters.

F. W. HVOSLEF: Friction loss through the heating unit is another effect which tends to reduce the flow of air through a convector. Flow of air through the heater is induced by the draft within the stack and as the draft is practically infinitesimal, this friction loss must be overcome before there is any air flow.

Tests of certain convectors show that after the stack is brought down to a certain minimum height, the flow of air practically ceases with the result that there is no heating effect from the convector.

Something that has been very confusing to me in my study of capacities of convectors and the manufacturers ratings thereof, is the apparently common belief that where cabinets or stacks are installed around heating units a better distribution of heat within the room results which justified the use of a smaller proportion of calculated heating surface than had heretofore been necessary with ordinary radiators.

The tests more recently published seem to indicate that a cast-iron radiator standing out in the room gives very nearly the same heating effect as a convector.

In view of these tests, is the heating engineer going to follow the lead of the manufacturers of convection heaters and assume that he can use 10 per cent or 20 per cent less cast-iron radiation than he has been in the habit of doing, or is he going to accept the so-called heating effect ratings which, as far as I can see from the data published to date, have no valid basis?

PROF. A. P. KRATZ: In order to get complete correlation and information on what convectors will do, both with reference to the correction factor and to the correlation between the cold room and the warm room tests, a great deal more work must be done because the present results give only trends.

The two things to which we might attribute the deviation are: the type of the convector, which is a rather intangible thing inherent in the unit itself, and the conditions established in the cold room which might not be the optimum conditions for that particular convector.

Tests in the cold room must be run under some established temperature conditions. The radiator must establish those conditions itself since it is limited entirely by the heat loss from the room. We were not certain but what that factor was affecting the performance or affecting the correlation between the two, and the curve that we plotted was plotted merely to serve as a straw in the wind, if you want to put it that way, to indicate whether that particular factor was having any bearing.

Our conclusions were that some particular factor was probably having a bearing and that therefore the tests in the cold room would have to be made under very much more closely specified conditions than the tests in the warm booth. Otherwise greater deviations were liable to occur in the cold room than in the warm room booth.

We tried what Mr. Hutzel mentioned, that is, putting a thermocouple beside the thermometer to find out how much difference in temperature would be read by the thermocouple and the thermometer. We found differences of 1 to 2.5 F in the trials that we made and in the same direction that Mr. Hutzel indicated. Therefore, undoubtedly part of the deviation could be attributed to that source. However, the corrections made for that would not bring the two sets of tests into complete correlation, and we feel that some other explanation must be offered and some other factors found for completely explaining the deviations.

We have not had any success with shielding thermometers against radiation. Shields were placed around them, but in many cases we found that the presence of the shield restricted the air flow over the thermometer and that the shield increased

the radiation effect received by the thermometer rather than decreased it. In order for a shield around a thermometer to effectively decrease the radiation effect, the shield should be mechanically ventilated.

Mr. Ewald brought out a point with reference to the cooling of the air in the cabinet reducing the stack effect. Actually we have found that the presence of the cold wall does not act that way because in a little while, after equilibrium is reached, the cold wall back of the radiator is no longer a cold wall but it warms up and serves as additional heating surface and offsets any loss obtained by cooling in the stack.

The question involving the heating effect in the rating of radiators is one that we are not in a position to answer at present. In this paper we have made no attempt to rate the radiators in terms of either condensation or of heating effect.

PROF. A. C. WILLARD: For 20 years the University of Illinois has been engaged in this study of performance of various types of heating and ventilating equipment and apparatus and this paper, which has just been presented, illustrates again the great importance to the engineer and to the equipment and apparatus manufacturer of conducting tests on the performance characteristics of radiators, fans, pumps, traps, anything you please, under the conditions of service. The unit under consideration must be given an opportunity to perform in precisely the same environment at the time the test is made as the environment when the unit is put into service. Laboratory tests are interesting and they often cause a great deal of discussion; they are also the source of a great deal of information; but to get the facts you have got to reproduce under the test conditions the environmental factors that affect the performance of the equipment or you will secure information that may be extremely misleading.

No. 939

LOSS OF HEAD IN COPPER PIPE AND FITTINGS

By F. E. Giesecke[1] (*MEMBER*) and W. H. Badgett[2] (*NON-MEMBER*),
College Station, Texas

This paper is the result of research conducted at the A. & M. College of Texas in co-operation with the A. S. H. V. E. Research Laboratory and the Associated Copper Tubing Manufacturers

THE purpose of this investigation was to determine the losses of head in copper pipe and fittings. Tests were conducted on ¾-in., 1-in., 1¼-in. and 1½-in. copper pipe, joined with couplings modified and installed as described in the following paragraphs, ¾-in., 1-in., 1¼-in. and 1½-in. copper elbows, and 1-in. copper tees. The water was maintained at a mean temperature of 84 F for all tests, varying not more than 1 deg from that mean. The room air temperature varied not more than 4 deg from that of the water.

Description of Apparatus and Procedure

The set-up used in these experiments is shown in Figs. 1, 2, 3 and 4. It consists essentially of an apparatus for securing a constant water temperature and a constant pressure head, a pipe line containing the pipes and fittings to be tested, and devices for measuring the velocity of the water in the pipes and the corresponding losses of head.

The water supplied for these tests came from the College mains and varied in temperature from time to time. To eliminate this variation, a heating device was constructed by means of which it was possible to control the temperature of the water within one degree of an established mean. This device consisted of a pipe coil, through which the water passed, immersed in a steam bath. By means of valves, a portion of the water was passed through this coil and raised to a higher temperature and then mixed in the proper proportion with the water coming directly from the mains so as to secure water of the desired temperature to be used in the tests.

Since the studies were conducted during the summer months when the water taken from the College mains had a temperature ranging from 70 F

[1] Director, Texas Engineering Experiment Station, Agricultural and Mechanical College of Texas, College Station, Texas.
[2] Research Assistant, Texas Engineering Experiment Station, Agricultural and Mechanical College of Texas, College Station, Texas.

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Milwaukee, Wis., June, 1932.

to 80 F, it was deemed advisable to use a mean temperature above 80 F in order to facilitate temperature control, and a mean temperature of 84 F was selected and maintained throughout the tests.

From the heating coil, the water passed into an elevated supply tank in which a constant head was maintained by means of an overflow pipe. From this supply tank the water flowed by gravity through a connecting pipe line into the pipes and fittings to be tested, and from there into a weighing tank. The rate of flow was controlled by means of gate valves placed on each side of the pipe and fittings tested.

The velocities were determined by weighing the discharge in tanks placed on a set of scales. By means of a stop watch, the time required for a pre-

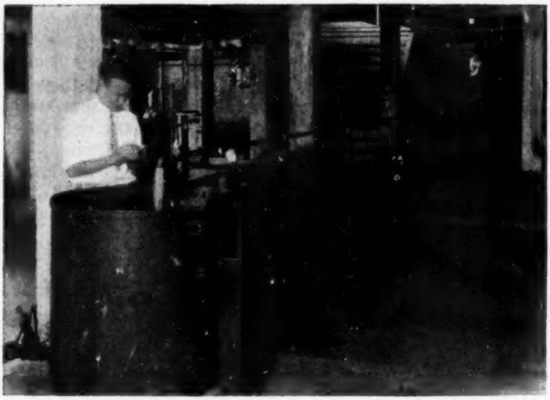


FIG. 1. DETERMINATION OF THE LOSS OF HEAD IN COPPER PIPE

determined quantity of water to be discharged was measured. The internal diameters of the pipes were obtained by means of calipers, and the results found to check within 0.001 in. with the factory rated diameters.

Loss of head was measured by means of piezometer rings placed at the beginning and end of the test sections and connected, by means of ¼ in. pipe and rubber tubing, to vertical manometer tubes. The three manometer tubes used had equal degrees of capillary action. The manometer scales could be read directly to the nearest millimeter, and, by interpolation, more accurately.

PIPE TESTS

For each size of pipe tested, 40 ft of pipe were set up with piezometer rings placed 5 ft from each end, making the length of each test section 30 ft. This pipe line consisted of 4 pipes 10 ft long and 3 couplings from which the separating projection had been removed in order that the two adjacent pipes were in direct contact with each other. (See Figs. 1 and 2.) Before assembling the 4 sections of pipe, the inside of each end was sandpapered to

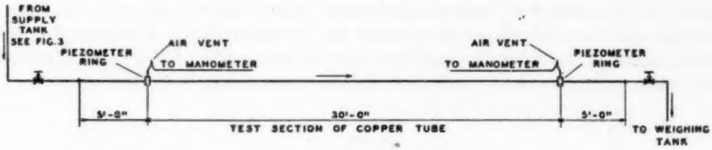


Fig. 2. Apparatus for Determining the Loss of Head in Copper Pipe

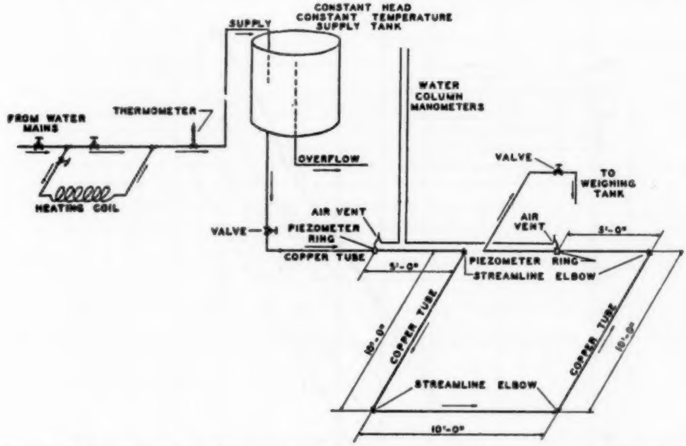


Fig. 3. Apparatus for Determining the Loss of Head in Copper Elbows

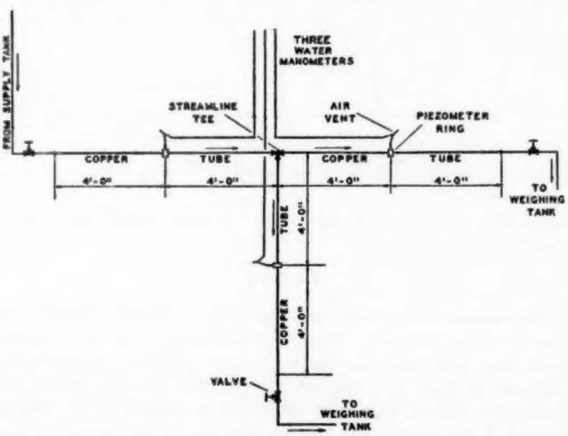


Fig. 4. Apparatus for Determining the Loss of Head in Copper Tees

remove any possible burr and the ends were placed tightly against each other. Extreme care was exercised in making the connections to eliminate the possibility of solder running through the joint to the inside of the pipe. Sections were taken apart for examination to make certain that none had entered. A

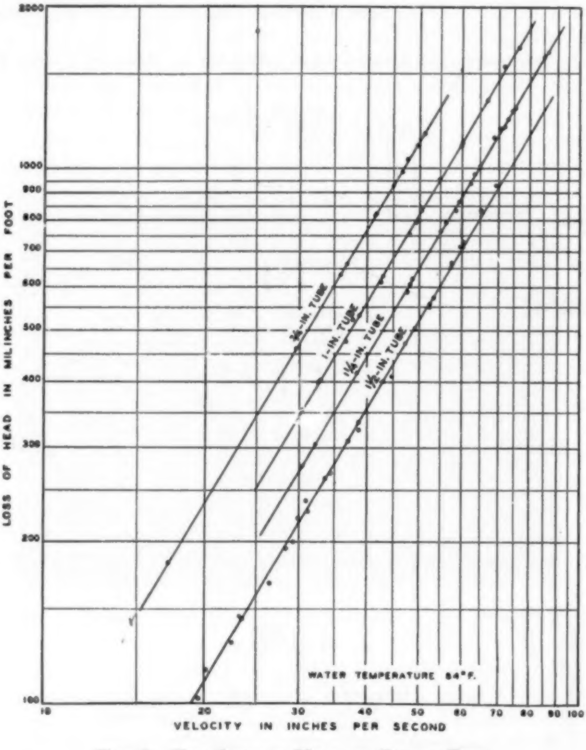


FIG. 5. THE LOSS OF HEAD IN COPPER PIPE

series of velocities was used on each size of pipe tested, and the results are shown in Fig. 5.

ELBOWS

After all laboratory tests were satisfactorily completed on the copper pipe, tests were made on the elbows. The couplings joining the 10-ft lengths of pipe were removed, the pipe ends carefully cleaned, and the couplings replaced by elbows as shown in Fig. 3. The set-up for each size of elbow consisted of 40 ft of pipe and 4 elbows connected in series as shown in the diagram.

The velocity of the water flowing in the pipe and the corresponding loss of head in the 40 ft of pipe and the 4 elbows were then determined for a

series of velocities. From these values the losses of head in 40 ft of pipe were deducted and the remainder divided by 4 to determine the loss of head in one copper elbow. The data for these tests are shown in Figs. 6, 7, 8 and 9. The results for the four sizes of elbows are shown collectively in Fig. 10.

TEES

The apparatus and procedure for determining the loss of head in the tees were the same as for determining the loss of head in cast-iron tees, described

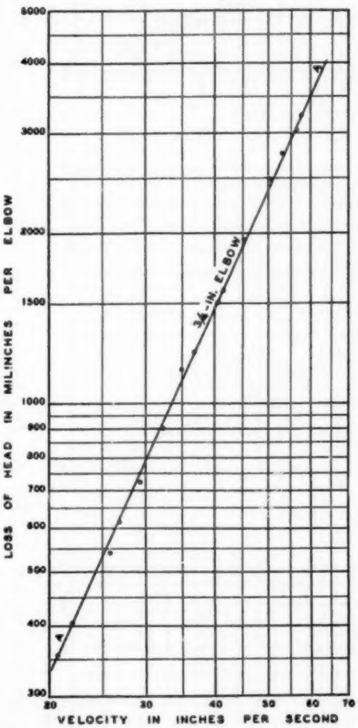


FIG. 6. THE LOSS OF HEAD IN ¾-IN. COPPER ELBOWS

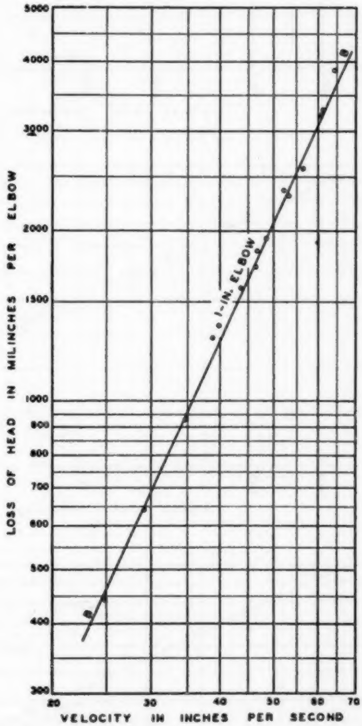


FIG. 7. THE LOSS OF HEAD IN 1-IN. COPPER ELBOWS

in the papers, Friction Heads in One-Inch Standard Cast-Iron Tees, and Supplementary Friction Heads in One-Inch Cast-Iron Tees.[8] Two set-ups were

[8] Friction Heads in One-Inch Standard Cast-Iron Tees, by F. E. Giesecke and W. H. Badgett (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931), and Supplementary Friction Heads in One-Inch Cast-Iron Tees, by F. E. Giesecke and W. H. Badgett (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

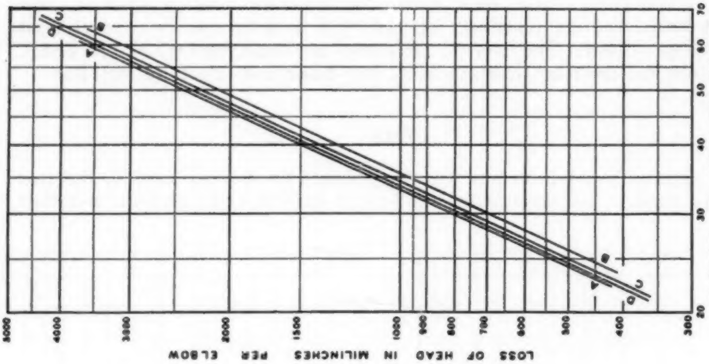


FIG. 10. THE LOSS OF HEAD IN COPPER ELBOWS, FOR THE FOUR SIZES TESTED, SHOWN COLLECTIVELY

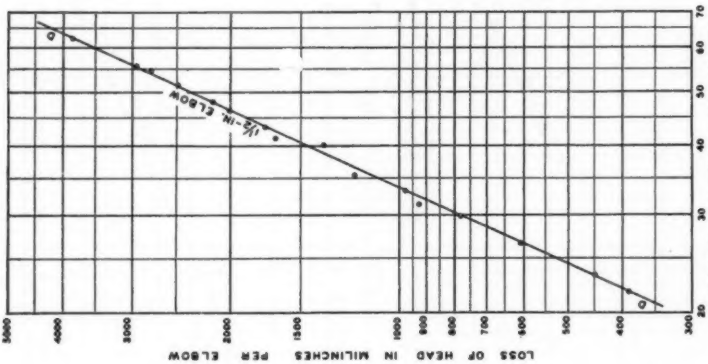


FIG. 9. THE LOSS OF HEAD IN 1½-IN. COPPER ELBOWS

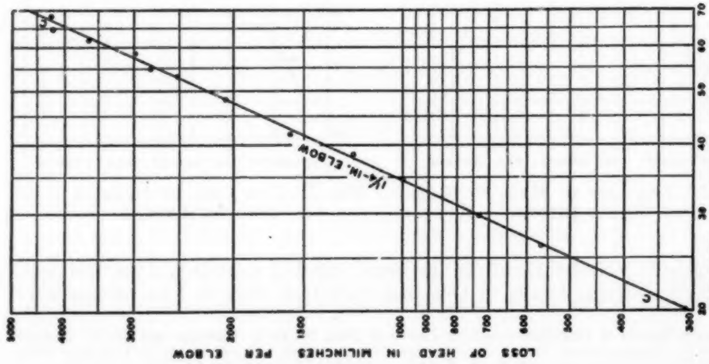


FIG. 8. THE LOSS OF HEAD IN 1¼-IN. COPPER ELBOWS

used, one with the water entering an end branch of the tee and a part of the water flowing out the side branch with the remainder continuing straight through the tee (Fig. 4). The data for these results are shown in Fig. 11. In the other set-up (Fig. 12), the water entered the middle branch of the tee and flowed out the two end branches. The data for the results in this case are given in Fig. 12.

When the water flows through the tee, as shown in the illustration of Fig. 11, and 70 per cent of the water entering at *A* is diverted through the outlet *C*,

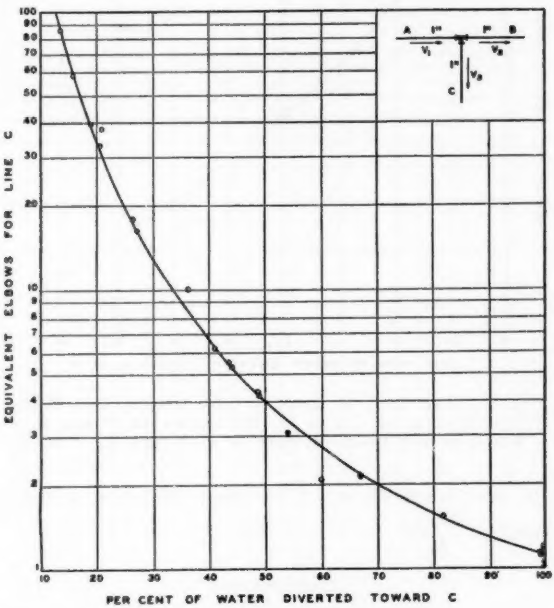


Fig. 11. The Loss of Head in 1-In. Copper Tees When Water Enters at *A* and a Portion Is Discharged at *C*, in Terms of the Friction Head of an Elbow at *C*

the loss of head in the portion so diverted is equivalent to the loss of head which would result if the water had flowed through two elbows instead of through the tee. Similarly, if 50 per cent is diverted, the loss of head is four elbow equivalents, as shown by the curve of Fig. 11.

When the water is flowing through the tee as shown by the illustration in Fig. 12 and 70 per cent of the water entering at *C* is diverted through the outlet *A*, the loss of head in the portion so diverted is 3 elbow equivalents. In this case, 30 per cent is diverted through the outlet *B* and the loss of head in that portion is about 16 elbow equivalents, as shown by the curve of Fig. 12.

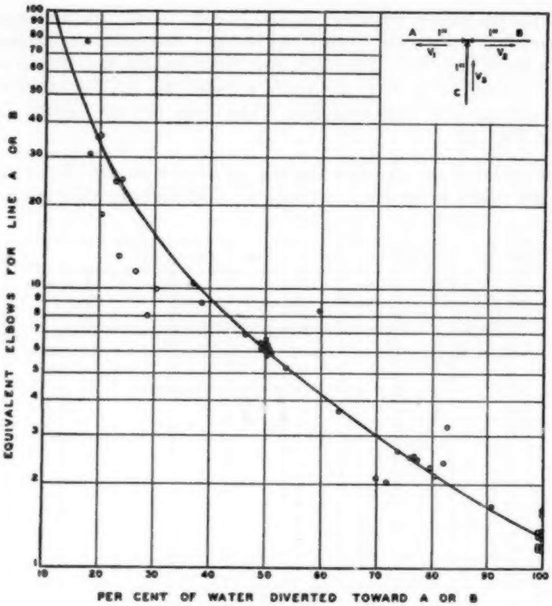


FIG. 12. THE LOSS OF HEAD IN 1-IN. COPPER TEES WHEN WATER ENTERS AT *C* AND IS DISCHARGED AT *A* AND *B*, IN TERMS OF THE FRICTION HEAD OF AN ELBOW AT *A* OR *B*

The loss of head in one elbow, *i.e.*, an elbow equivalent, as used in Figs. 11 and 12, is equal to $\frac{0.7v^2}{2g}$.

CONCLUSIONS

The loss of head in copper pipe for 84 F water may be expressed by the formula:

$$h = 1.16\frac{v^{1.7}}{d^{1.22}} \tag{1}$$

where

h = the loss of head in milinches of water per foot of pipe

v = the velocity of the water flowing in the pipe in *inches* per second, and

d = the actual inside diameter of the pipe in inches, and varies from ¾ in. to 1½ in.

The loss of head in a copper elbow may be expressed, approximately, by the formula

$$h = 0.7\frac{v^2}{2g} \tag{2}$$

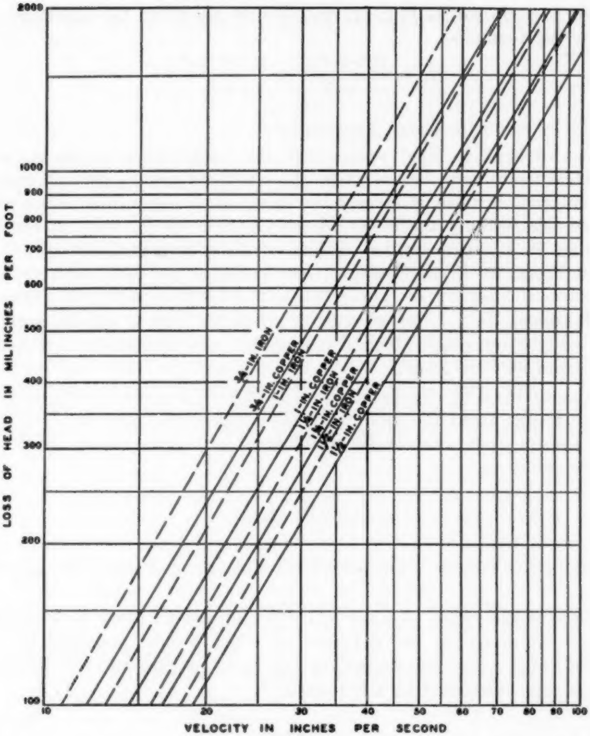


FIG. 13. LOSS OF HEAD IN VARIOUS SIZES OF IRON AND COPPER PIPE

where

h = the loss of head in milinches of water

v = the velocity, in inches per second, of the water in the pipe fitting the elbow

g = the acceleration due to gravity (386 in. per second each second)

The loss of head in a copper tee may be expressed, approximately, by the formula

$$N_e = \frac{0.7\,(v_1^2 + v_3^2)}{v_3^2} \tag{3}$$

where (for the case shown in Fig. 11)

N_e = the number of elbows which would cause the same loss of head as the tee when the velocity of the water in the connecting pipe is v_3

v_1 = the velocity of the water in the pipe entering the tee

v_3 = the velocity of the water in the pipe discharging from the tee at right angles to v_1

For the case shown in Fig. 12, the loss of head can be expressed, approximately, by the formula

$$N_e = \frac{v_3^2 + v_1^2}{v_1^2} \text{ or } \frac{v_3^2 + v_2^2}{v_2^2} \qquad (4)$$

Effect of Variation in Water Temperature

The loss of head in pipes varies with the temperature of the water because the temperature affects the viscosity, the viscosity affects the internal friction, and the internal friction is largely the cause of the loss of head in pipes. The method of determining the effect of temperature on the loss of head in pipes is described in a paper entitled, Effect of Temperature Upon the Friction of Water in Pipes, by F. E. Giesecke,[4] presented at the 31st Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Boston, Mass., January, 1925. For a velocity of 3 fps and a 1¼-in. pipe, the loss of head will be increased about 5 per cent if the temperature of the water is lowered from 84 F to 70 F.

The loss of head in elbows and tees is caused largely by the loss of kinetic energy resulting from the change in the direction of the flow of water and, for that reason, is practically independent of ordinary changes in the temperature of the water.

APPENDIX

COMPARISON OF COPPER PIPE AND FITTINGS WITH IRON PIPE AND FITTINGS

To compare the losses of head in copper pipe with those in iron pipe of the same nominal diameter, the diagram of Fig. 13 was prepared by using the values for copper pipe determined in this investigation and the values for new clean iron pipe calculated by the formula [5]

$$h = 0.00685 \frac{v^{1.77}}{d^{1.275}} \qquad (5)$$

The loss of head in a standard cast-iron elbow is practically equal to $\frac{v^2}{2g}$; the loss of head in a copper elbow is about $\frac{0.7v^2}{2g}$, provided the internal diameter of the elbow is equal to that of the pipe connected to it and provided the two are joined so that there is no sudden enlargement or contraction in the channel through which this water flows, as was the case with the elbow and pipe tested in this investigation.

DISCUSSION

H. F. HUTZEL: I note in Fig. 13 that the loss in standard pipe is somewhat greater than in copper pipe of the same nominal pipe size. In checking formulae 1 and 5 which express the loss of head in milinches of water per foot of pipe, for

[4] See A. S. H. V. E. TRANSACTIONS, Vol. 31, 1925.
[5] See *University of Texas Bulletin* No. 1759, p. 3.

both copper and steel respectively, I cannot reconcile the formulae with the results shown in your Fig. 13.

For example a 1-in. pipe having an inner transverse area of 1.48 sq in., and assuming a velocity of 20 in. per second, H for the copper pipe figures out to be 178 milinches and for the steel pipe as per formula 5—1.29 milinches. The H as calculated by formula 1 checks with the head loss represented on chart Fig. 13, but H for the steel pipe does not.

Is it possible that the coefficient in formula 5 is in error or does not the H in formula 5 represent the loss of head milinches?

With reference to the loss in elbows and tees, the paper does not explain whether these were cast fittings and whether the elbows were of short radius of a pattern similar to ordinary cast iron tees. I gather from the description that these fittings are similar to sanitary fittings.

JOSEPH LEGRAND: The experimental work reported by Professor Giesecke and Mr. Badgett upon the friction losses of pipes and fittings covers a subject upon which there has long been needed more precise engineering information. While it is essential primarily to obtain basic precision in the determination of losses in new pipe and fittings, it is still more desirable that engineers be supplied with more exact data concerning the effects of time and various waters upon these losses than is at present readily obtainable.

The variation of climate, waters and time are appreciable factors in friction losses. Such terms as "a growth of slime," "heavy incrustation," "a layer of rust," and tuberculation are far from exact terms and subject to widely different interpretation. An investigation into this subject upon a fixed basis would be of value.

More data are desirable upon kinetic losses brought about by changes in direction of the flow, the action of various valves and fittings and the resultant action of fitting combinations, such, for example, as an elbow immediately following the discharge of a globe valve, a pair of elbows connected together at right angles, a valve located adjacent to the discharge of the branch of a tee and many other combinations. It is easily conceivable that some of these combinations occasion losses which could be prevented by the simple expedient of a slight increase in distance between the fittings.

In connection with the losses in head resulting from loss of kinetic energy, it is suggested that these losses be investigated in fire hydrants and the connections between the hydrants and the high pressure mains serving them.

In Fig. 4 of the paper the authors designate a tee as "streamline tee," and in Fig. 3 elbows are similarly designated. Considering the publication of the writer's paper [6] upon the subject of streamline elbows, tees and valves, the writer feels that a misinterpretation may result. The fittings referred to in Giesecke and Badgett's paper have no relation to the fittings described in the writer's paper. The elbows and tees used by Giesecke and Badgett were regular constant area fittings, termed "streamline fittings" by the manufacturer, due to the fact that a smooth connection between the fitting and the pipe is obtainable. The use of the term "streamline" should be limited to non-turbulent flow, or flow in which the velocity is below the critical velocity of the fluid.

The writer understands from Prof. Giesecke that some work is now being done under his direction upon the loss of head in globe valves and other fittings. Similar work is being undertaken, with the addition of an investigation into the streamline anti-turbulent valves described by the writer in his paper upon that subject, by Prof. J. C. Peebles and the writer at Armour Institute of Technology. It is hoped that the results of these experiments will prove of constructive value.

[6] See Methods of Reducing Friction Heads in Pipe Fittings, A.S.H.V.E. JOURNAL SECTION, *Heating, Piping and Air Conditioning*, May, 1932.

M. BARRY WATSON: The paper is most timely, for with copper at its present low price level, there is no doubt but that this material will be extensively used in hot water heating systems, where a knowledge of its hydraulic characteristics are essential for economic designing.

The technique adopted in these tests appears, from the close coincidence of the test results with the average curves, to be all that could be desired.

The writer had occasion during the past year or so to design a number of gravity hot water heating systems using this type of copper pipe and fittings, and was able to locate very meagre authoritative data on which to base such designs. Possibly the simplest presentation of rational hydraulic data can be found in Walker and Crocker, *Piping Handbook*. As a check on the basic values there presented, I have made the following check computations to compare with the author's results.

The general formula for pressure loss in a pipe for water is developed in the form,

$$Pl = \frac{0.323\, f\, v^2}{d}$$

in which,

Pl = Pressure drop per foot of length in pounds per square inch.
v = velocity of water in feet per sec.
d = inside diameter of pipe in inches.
f = friction factor.

The following formula for determining the value of the friction factor water in drawn copper piping is taken from Wilson, McAdams and Seltzer (*Jl. of Ind. & Eng. Chem.*, p. 114, Feb. 1922)

$$f = 0.0018 + 0.00662\left(\frac{z}{d\,v}\right)^{0.355}$$

in which z is the absolute viscosity of the fluid in centipoises, relative to water at 68 F.

Taking as a check example, a 1¼ in. copper pipe (1.375 in O.D. with 0.042 in. wall) having an inside diameter of 1.291 in., with water at 84 F and velocity of 5 ft per second. $z = 0.82$; $d = 1.291$; $v = 5$; and f becomes 0.00498 and Pl becomes 0.03122 lb per square inch per foot of pipe, or a loss of head of about 720 milinches per foot. The authors' formula gives for these conditions 895.3 milinches per foot of pipe. Peculiarly the experimental curves for the next larger pipe size shows almost exactly 720. Obviously the coefficients in the formulae which I used are inconsistent with the facts.

The operation of the systems which I designed on the above basis, while not yet fully analysed, has demonstrated the total friction in the system is less than expected, since the drop in water temperatures through the radiators at full load is only about three quarters of the calculated drop, indicating faster circulation than expected. This result would now appear to be due to an excessive allowance for fittings, etc., which over-corrected the inadequate piping factors used.

The frictional data presented relative to this particular design of elbow is particularly valuable.

With regard to the data developed on tees; and this criticism is also applicable to the authors' earlier tests on cast-iron tees,[7] it is my experience that where one pipe divides its flow more or less evenly between two branches, a rational design frequently makes the branch pipes one size smaller than the main, thus maintaining nearly constant velocities, while the authors have considered only the condition where the branch pipes are the same size as the main. There are of course some good

[7] See Friction Heads in One-Inch Standard Cast-Iron Tees, A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, and Supplementary Friction Heads in One-Inch Cast-Iron Tees, A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932.

reasons for decreasing velocities progressively from the boiler toward the ends of the circuits, but some data on division of water at more constant velocities would be more valuable.

From elementary hydraulic reasoning one would suspect that in dividing the flow of say a 1 in. copper pipe equally into two ¾ in. pipes, in which case the average velocity will be decreased about 15 per cent; the loss of velocity head, and probably total loss in the tee, would be considerably less than where the velocity is reduced about 50 per cent as in dividing the flow equally between two pipes the same size as the supply.

The authors did not state whether they tested the friction in tees corresponding to the return piping of a heating system, *i.e.*, where water is combining instead of dividing. It would be interesting to know whether such friction is at variance with the arrangement corresponding to supply piping.

The data published on bull-head tee connections is very useful, as with these pipes and fittings a tee can be connected in this way at the top of a riser and the small clearance in the tees is sufficient to allow the run-outs to be properly graded, which can not be satisfactorily done in the case of iron pipe, thus eliminating one fitting which is worth considering in the cost of a job where brass fittings are used.

If the authors had included in their paper more complete data on the effects of temperature variations, it would have been of value, especially to the younger members of the Society whose libraries do not extend back to the reference given, and who have not available, or are unfamiliar with the application of, basic data on this effect.

I wish to heartily congratulate the authors for publishing the data which they have obtained up to date, as it fills an immediate need.

CHAS. A. HILL (WRITTEN): Engineers will welcome the information given by Prof. Giesecke and Mr. Badgett since they give some definite facts concerning a new development in piping.

The value of brass or copper pipe on all hot water lines has long been recognized, but cost has been a factor that has restricted the use of these materials in the iron pipe sizes which have been generally used.

The recent introduction of soldered fittings and hard drawn copper pipe (or tubing) of much less thickness than iron sized pipe has materially reduced the cost of copper pipe jobs in all sizes up to 6 in. that its use has grown rapidly, not only for plumbing, hot water heating, and returns on steam heating, but also for mains, risers and branches in steam heating.

It has been customary to reduce pipe sizes when substituting brass or copper for steel pipe. This has been done mainly on the assumption that there will be no reduction in size by accumulation of rust over a period of years.

The sizes called for in the pipe tables usually give the figure fully 20 per cent over computed requirements. The more or less rough interior of a pipe and its certain rusting justifies this.

When iron size brass or copper is used with the customary screwed pipe fittings, friction losses are much reduced in the pipe but the loss of head in the fittings is not noticeably helped. One distinctive feature of the soldered fittings is that the cross section of the opening remains the same as in the pipe; in other words, there is no increase in diameter of stream through the fitting.

Prof. Giesecke has shown that the loss of head through fittings of this type is but seven-tenths of the loss on screwed fittings. He has also shown that copper pipe will permit an increased velocity of from 10 to 20 per cent before the loss of head equals that in similar sizes of iron pipe.

These conclusions will give a basis for the intelligent calculation of sizes when using hard drawn copper pipe and soldered fittings.

Prof. Giesecke's paper in connection with the data[8] developed by the Society's Research Laboratory gives valuable and definite information needed by the many engineers who have recognized the value and economy of this new development in copper pipe and its fittings.

W. H. BADGETT: In reply to Mr. Hutzel's discussion, Equation 5 is correct, but the units are not the same as those of Equation 1. The loss of head, h, in Equation 5 is expressed in feet of water, v is the velocity in feet per second, and d is the inside diameter of the pipe in inches.

The fittings used in these tests were of short radius of a pattern similar to ordinary cast iron fittings with the exceptions that the inside of the fittings were of constant cross-sectional area and that soldered rather than threaded joints were used.

[8] Heat Emission from Iron and Copper Pipe, by F. C. Houghten and Carl Gutberlet, A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932.

No. 940

AUTOMATIC GAS BURNERS

By C. George Segeler,[1] New York, N. Y.

NON-MEMBER

THE application of gas can be made through two general methods, namely (1) by the use of equipment specially designed for gas burning, and (2) by installing gas burners in existing equipment intended for some other fuel. Considerable information has been published on the use of gas with specially-designed equipment, but there has been comparatively little on the automatic gas burner used for converting existing equipment to gas operation.

The units for accomplishing this purpose are termed *conversion burners.* Since these burners are really gas burners fitted with the same equipment and controls used with boilers or furnaces designed for gas operation, it is to be regretted that the term *conversion burner* has found acceptance.

The published papers on automatic gas burners have generally been written from the standpoint of the operating cost. Although it is true that greater care is necessary in the selection of conversion burners where the gas cost is high, it is also true that automatic gas burners can and are being successfully used in territories in which gas-designed equipment is used. In the past, statements to the effect that automatic gas burners could be made as efficient as gas-designed equipment have often been questioned, but as the author will endeavor to show later, this is sometimes the case if sufficient care is exercised in selecting the jobs converted and proper installation and adjustment of the burner is made afterwards.

Where automatic gas burners are to be used in existing buildings, the installed equipment must be adequate, it must not be too old, and it must be of a type for which a fairly efficient conversion can be made.

Although there are exceptions, small boilers usually are less satisfactory than large boilers for conversion to gas operation. This is particularly true of small round boilers without intermediate sections. Such a statement in no way reflects upon any manufacturer of coal-fired boilers. It does not in-

[1] Industrial Engineer, *American Gas Association.*

Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Milwaukee, Wis., June, 1932.

dicate that such a boiler is not efficient when used with coal, but merely that the construction is such that it is not well adapted for gas.

Coal-fired, warm-air furnaces can be divided into three classes, namely (1) self-cleaning dome type, (2) radiator dome type and (3) downdraft type. The latter two types can be changed to automatic gas operation satisfactorily, but considerable care must be exercised in converting the first type.

Lest any misunderstanding arise from the foregoing reference to the difficulties attendant upon adapting the automatic gas burner to the small coal boiler, it should be noted that there are also other factors than size which may limit the use of gas burners. Space will not permit enumerating all of them, but the principal points are as follows:

1. There is a difference in boiler construction between units intended for anthracite and bituminous coal, principally based on draft consideration.

2. Age of the boiler, especially when hard water is used.

3. Capacity of the boiler compared to the heating requirements of the house.

4. The chimney may be a factor, since flues too small for good coal operation may often be entirely satisfactory for oil or gas burning. Outside exposed chimneys with unlined flues are liable to be unsatisfactory.

Having more or less defined the field of usefulness of the automatic gas burner, the differences between the individual burners themselves can be considered. Automatic gas burners may be simply classified into the following 3 groups: (1) elementary atmospheric burners, (2) burners with secondary air control, and (3) burners with air blowers.

ELEMENTARY ATMOSPHERIC BURNERS

The simplest burners consist of plain drilled pipe set into the firebox of the coal boiler. No attempt is made to control the amount of excess air which enters the furnace and no refractories are used. Burners of this class are made by many local concerns throughout the natural gas territory. They are obviously inefficient and undesirable from a service angle.

Special shapes of atmospheric type burners are also used, such as the single port burners which are set into the fire door and play a large volume of flame from a single port. The nozzle-mixing multi-port burners, also of this class, are widely used not only for domestic installations but for the conversion of the larger size boilers. For the latter, this class has certain specific advantages in simplicity and long life, the details of which need not be considered in this paper.

BURNERS WITH SECONDARY AIR CONTROL

Because many of the installations made with elementary atmospheric burners yielded rather low efficiencies, secondary air-controlled types of automatic gas burners were developed. These are arranged to admit all air for combustion through a sheet metal duct surrounding the burner. An air control damper opens and closes with the position of the gas valve, thereby controlling the secondary air during the *on* period, and shutting off the air flow during the *off* periods.

BURNERS WITH AIR BLOWERS

The third class of burners consists of those using an air blower for the purpose of securing improved combustion and better control. These units do not operate with 100 per cent primary air mixed with the gas by the blower. This may be surprising to many, since it is frequently assumed that the blower system would inject all of the required air with the gas. This is not usually done because of the nature of gas combustion. In order to obtain silent combustion and freedom from flashback, it is desirable in house heating equipment to mix about 60 per cent of the air required as primary air.

Practically all automatic burners are designed to locate the flames in close proximity to the heating surface. This has been found to give a relatively high efficiency, and makes the lighting of the burners very positive. However, there are certain exceptions to this construction, notably in the elementary atmospheric types of burners and in the burners designed to operate with special refractory shapes in the center of the firebox.

There is another type of burner which does not fall exactly into any of the classifications mentioned. In this class there is no air injection external to the boiler. The burners are designed from small bunsen-type tubes, the air and gas mixing taking place within the combustion space itself.

With so many different types and styles available, the question arises as to which should be selected. There seem to be 4 principal factors involved. *First,* that type of burner should be selected which would require the smallest investment consistent with safety, efficiency and satisfactory service. *Second,* the burner selected should cause the least possible change in existing equipment, so that the owner would not feel that he was permanently bound to gas fuel and so that the working principle of the unit would be most readily understood. *Third,* the unit should possess a neat appearance when installed and should have a low cost for installation and above all, a low cost for servicing. *Fourth,* the unit should be such that the owner receives an efficient and safe device.

INSTALLATION OF AUTOMATIC GAS BURNERS

Considerable experience is required to satisfactorily install an automatic gas burner in existing equipment. With gas-designed equipment, it is merely necessary to follow instructions for assembly to be sure of good operation. This is well within the ability of any capable mechanic. For the installation of automatic burners, on the other hand, it is not only a question of following certain instruction rules, but it is also necessary to have a mechanic with sufficient knowledge of gas combustion and heating methods to be able to make the necessary changes in the original equipment so that with the burner it will make an efficient and safe installation.

The *American Gas Association* Testing Laboratory Seal of Approval which is awarded appliances that meet the specifications of the *A. G. A.* General Requirements Committee [2] cannot be extended to cover automatic gas burners because, although the gas-designed unit can be accurately assembled by any

[2] This committee consists of representatives of U. S. Bureau of Standards, U. S. Bureau of Mines, U. S. Public Health Service, U. S. Bureau of Home Economics, *American Home Economics Association,* Underwriters Laboratories, *Heating and Piping Contractors National Association, National Association of Master Plumbers, Canadian Gas Association* and gas company and manufacturer members of the *A. G. A.*

competent mechanic so that it will comply with the requirements under which it was tested for approval, the automatic gas burner depends upon the skill of the installer. Consequently, it would be impossible to issue a Seal of Approval, which bears the label, *Complies with basic national standards for safety.*

In order that the automatic gas burner might be given consideration, the *A. G. A.* General Requirements Committee has approved the issuance of the Installation Requirements for Conversion Burners. These are requirements which, when followed, will make the finished installation comparable with gas-designed equipment as far as it is possible to do so with this equipment. These requirements are divided into several parts, covering the inspection and preparation of the existing heating plant, with particular reference to chimney flues and proper size of flue pipe. They provide that no baffles shall be installed in existing equipment which will interfere with proper combustion of gas, and recommend that the section of flue pipe between the outlet of the appliance and the draft hood shall be sized on the basis of 1 sq in. of area for each 7,500 Btu per hour input into the unit. The second part deals with the actual installation of burners and controls. In general, the same type and number of controls are needed for automatic gas burners as for gas-designed equipment. This includes pilots, main shut-off valves, gas pressure regulator, necessary limiting devices, etc. The third part deals with the gas piping and meter arrangement, and a final section is devoted to inspections and test. This material is available on application from the *American Gas Association.*

The Requirements represent minimum standards for the installation of automatic gas burners. They have been prepared in the interest of the public, to ensure safe and economical utilization of gas. Naturally, there are companies whose practices will go beyond these minimum standards and will thus be able to reach the highest levels of efficiency and good service. Most of the recommendations in the Installation Requirements are sufficiently definite that they require practically no further interpretation. There are some which have been deliberately left general, because of variations in local practice.

Most important of these is the method for sizing automatic gas burners to fit a given installation. The *A. G. A.* practice is based on first estimating the hourly heat loss of the house, and then providing a sufficiently high heat input per hour to the unit to take care of the following 3 points: (1) Efficiency of the unit; (2) Heat lost from risers, piping and leaders and thus not delivered to the rooms, and (3) Extra capacity for quick starting under thermostatic control.

For warm air furnaces, it is assumed that 90 per cent of the heat delivered to the heat distributing pipes is finally brought into the rooms of the house; provided that where leaders pass through unexcavated spaces, they are insulated and that the area of the air passages through the furnace is at least equal to the combined area of the take-off pipes.

Combining the probable efficiency of the automatic gas burner with these requirements, the input for warm air furnace conversions can be based on the following rule: *Multiply the hourly Btu loss of the house by* 1.56. *The answer will be the required hourly input for automatic burners for warm-air furnaces.*

The *A. G. A.* Requirements for gas-designed, warm-air furnaces call for gas-tightness between the heating section and the air passages. A similar criterion

should be applied to existing warm air furnaces before they are accepted for conversion to gas firing. If any doubt exists about the tightness of the joints in the heating surfaces, it should not be difficult to make a wintergreen or smudge test to make certain that the furnace joints are actually tight.

Boilers are sized in a similar manner. The *A. G. A.* long ago adopted a standard practice of sizing boilers up to 500 sq ft of installed equivalent heating surface (radiation) in the house on a basis of 56 per cent more boiler capacity than standing heating surface. The excess capacity decreases up to 4,000 sq ft of installed heating surface for which the figure is 40 per cent. In dealing with automatic gas burners, it has been suggested that a slightly smaller excess capacity can be satisfactorily used. The sizing can be done by multiplying the hourly heat loss of the house by the factors given in Table 1.

In converting large boilers it is advisable to work out the details of the installation rather than to depend on tabular factors for estimating the gas

Table 1. Factors for Sizing Boilers

Installed Heating Surface (Square Feet)		Factor	For Calculating Required Gas Input, Multiply Hourly Btu Loss by (75% Efficiency Assumed)
Steam	Hot Water		
200	350	1.58	2.11
250	400	1.57	2.09
300	470	1.53	2.04
400	640	1.50	2.00
500	800	1.46	1.95
600	1000	1.42	1.90
700	1100	1.38	1.83

required. Continuously-operating boilers, boilers for factories, clubs, etc., may have special requirements for quick pick-up, or the piping arrangements may be such that empirical factors are not as satisfactory. However, it is not possible to cover the details of such situations in the short space of this paper. The general rule can be applied that the excess capacity may be reduced as the boiler size increases.

It has been suggested that the proper operating efficiency for automatic gas burners be obtained by depending upon a flue collar or properly-sized flue outlet to provide the necessary baffling effect rather than using baffles or restrictions in the flue passages of the equipment. The safe basis for operation is 7,500 Btu input per hour per square inch of flue opening, and an oxygen content in the flue gases of not less than 4 per cent nor more than 10 per cent. Such a requirement does not necessarily provide the lowest obtainable flue temperatures. It should be readily possible, however, to bring these down to between 250 F to 350 F for steam boilers.

These are relatively broad requirements and some, interested in securing higher thermal efficiencies, would want to impose further restrictions. For example, by using a smaller flue opening and reducing the oxygen content of the flue gases to the minimum figure, the thermal efficiency can be increased slightly. According to the principles of operating gas appliances, there must be no carbon monoxide in the flue gases at any time. An ordinary flue gas analysis by the Orsat method will not be sufficiently accurate to use as a test

of compliance with the requirements for absence of carbon monoxide. In coal or oil-burning appliances, where unbroken flue connections are used, it would not be practical, and probably would be unnecessary, to attempt to restrict the carbon monoxide content of the flue gases to the same degree required for gas-burning appliances. The limit of carbon monoxide permissible for gas-designed central heating appliances is not more than 0.04 per cent in an air-free sample of the flue products. There are portable types of carbon monoxide indicators and detectors which give very quick determinations of carbon monoxide without going through the complications of an Orsat analysis. By such methods, efficiencies can be brought to high levels. In fact, under the conditions of a series of tests run by a well-known university laboratory, the efficiency of converted systems was found to be on a par with that of gas-designed equipment.

In order to obtain additional data along the lines developed by the university laboratory already referred to, an investigation was started among the larger

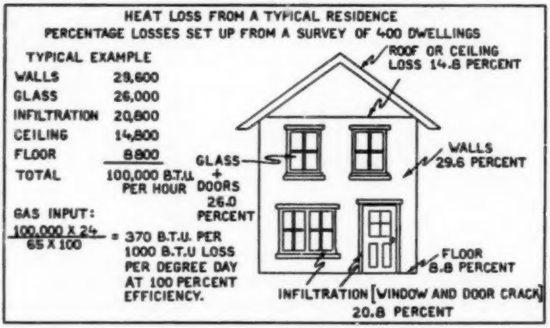


FIG. 1. RATIO OF OUTPUT TO INPUT

gas companies, obtaining from each a list of typical steam and hot water installations. This list, although a random selection, was picked to have approximately the same number and size installations in the following 4 classes: (1) Gas-designed hot water boilers; (2) Converted hot water boilers; (3) Gas-designed steam boilers, and (4) Converted steam boilers.

Thirty-two gas companies participated in sending in data, and more than 700 jobs were listed. A very excellent coverage of the field was thus obtained.

The data have been reported on the basis of Btu of gas required per degree day per 1,000 Btu hourly heat loss from the building. Theoretically, a unit figure can readily be calculated corresponding to 100 per cent thermal efficiency as follows:

Assume that a house having an hourly heat loss of 100,000 Btu, designed for an outside temperature of 0 F and an inside temperature of 70 F, were operated continuously through a period of zero days. That is to say, the house would lose

$$100{,}000 \times 24 = 2{,}400{,}000 \text{ Btu}$$

each day. On a zero day there are 65-deg days; consequently, the amount of fuel required to heat this house on a 100 per cent efficiency basis would be 370 Btu per degree day per 1000 Btu hourly loss from the house.

Fig. 1 illustrates the ratio of output to input of houses according to the methods used in this paper. The typical example tabulated at the left of the illustration assumes a house having a total heat loss of 100,000 Btu per hour, and divides this into losses from infiltration through glass and doors, walls, roof and floor. On the right of the illustration are tabulated the results of a survey of gas-fired houses made in 1930 by the author in an attempt to ascertain the percentage of heat lost through the various parts of the house. The houses were measured by the methods given in the A. S. H. V. E. GUIDE, and the percentages tabulated and averaged. The input in Btu delivered by the gas is given for various seasonal efficiencies on the basis of units of 1000 Btu loss per hour from the building per degree day.

The actual results obtained from the present survey of 700 private residences averaged as follows:

Type of Installation	*Btu per degree day per 1000 Btu hourly heat loss from the house*
Gas-designed hot water boilers	404
Gas-designed steam boilers	445
Converted hot water boilers	452
Converted steam boilers	503

These figures show overall seasonal heating efficiencies from a maximum of 91.6 per cent to a minimum of 73.5 per cent. They also show that the overall results of hot water equipment are 10 per cent better than steam equipment and that the difference between the average efficiency of all gas-designed versus converted equipment was about 12 per cent in favor of the gas-designed equipment.

Fig. 2 shows the relationship between the size of the installation and the average gas consumption, expressed in Btu of gas required for each 1,000 Btu hourly heat loss from the building, figured in accordance with the methods given in the A. S. H. V. E. GUIDE, per degree day. This figure refers to the results obtained for gas-designed hot water boilers. These data are all for private residences. As the size of the job increases the unit gas requirement seems to decrease.

Fig. 3 is a similar plot based on gas-designed steam boilers. Fig. 4 is another similar plot based on converted hot water boilers.

In all of these 3 figures, it appears that larger installations were more efficient than smaller ones. Perhaps there is some justification for this interpretation, but there are some other points to consider. For the moment, it may be assumed that these charts represent the actual averages of a number of installations in the various size groups plotted.

When the lists of jobs were studied in detail in any one size class, it was found that individual installations showed extremely wide differences in efficiency. It showed that there was a small group of gas-designed jobs that were better than the best of the conversions and it also showed that a small group of converted installations was less efficient than the least efficient of

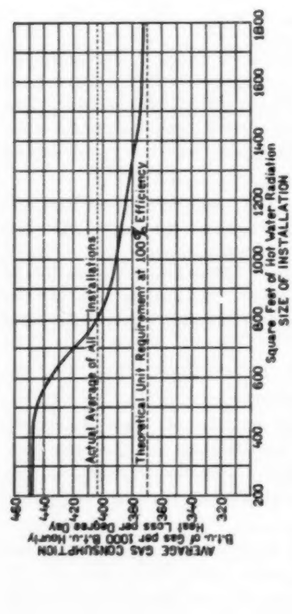


FIG. 2. UNIT GAS REQUIREMENTS FOR HEATING RESIDENCES WITH GAS-DESIGNED HOT WATER BOILERS

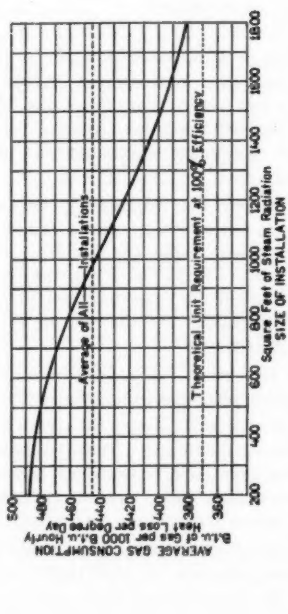


FIG. 3. UNIT GAS REQUIREMENTS FOR HEATING RESIDENCES WITH GAS-DESIGNED STEAM BOILERS

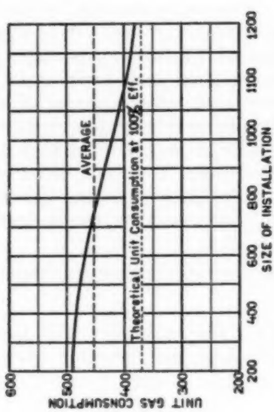


FIG. 4. UNIT GAS REQUIREMENTS FOR HEATING RESIDENCES WITH AUTOMATIC GAS BURNERS IN WATER BOILERS

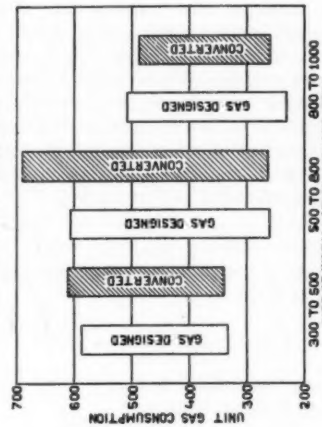


FIG. 5. RANGE OF UNIT GAS CONSUMPTIONS FOR GAS-DESIGNED AND CONVERTED HOT WATER BOILERS

the gas-designed equipment. The majority of the jobs was on a par as far as efficiency is concerned; the 12 per cent difference in average efficiency was largely caused by the installations which fell into the highest and lowest groups.

Fig. 5 illustrates the range of values obtained in this study. It can readily be seen how the averages worked out favorably to the gas-designed equipment even though there was a large number of installations in each class which yielded overlapping unit figures. Eliminating freaks, the worst of the converted installations averaged about 558 Btu per degree day per 1,000 Btu hourly loss from the building, as contrasted with 525 for the worst of the gas-designed.

The extraordinary part of these data was the unit figures obtained from the best of the installations. The best of the gas-designed jobs show about 250 Btu per degree day per 1,000 Btu hourly heat loss from the building, and the best of the converted installations about 274 Btu. Taking the two unit figures just mentioned as an example, they indicate that the overall seasonal thermal efficiencies were 140 per cent and 150 per cent respectively. In other words, the theoretical heat losses, calculated according to recognized principles, and making due allowance for every factor which was known to be involved, were found to be one and one-half times as great as the heat input furnished by the gas.

The practice followed by the engineers of the gas companies who contributed the data used in this paper is to draw up complete data sheets showing the hourly heat loss of each house. These calculations are made strictly in accordance with the methods developed by the Society and published in THE GUIDE. The fuel consumption is metered separately from the other uses for gas in these houses, and it is reasonable to assume that the meter accuracy is better than $\pm$ 2 per cent. The Btu value of gas, although it fluctuates somewhat, is averaged by the gas companies and it was these monthly average figures of Btu input which were correlated with the heat loss of the house. It was expected that a reasonably close check could be obtained between the input in gas Btu and the seasonal heat loss of the house.

So much was expected, but actually the results turned out as has been indicated, namely, that there was no very close check between the input and the output of houses, and that it was unwise to attempt to employ heat loss calculations on buildings as a basis for estimating the seasonal efficiency of various heating systems.

If a close check had been obtained between the maximum and minimum unit figures, or between the input and output estimates of houses, it would point out the value and desirability of refining still further the excellent methods for calculating and estimating heat losses and heat requirements as outlined in THE GUIDE and in the many valuable research papers published by the Society based on results obtained at the A. S. H. V. E. Research Laboratory and at the cooperating institutions. The results obtained did not show a close check, but on the contrary a wide divergence; high and low values were as much as 2 to 1 in relation to one another. Seasonal efficiencies figured out to theoretical results of better than 100 per cent. These factors have raised some question in the minds of the gas engineers on the advisability of carrying out too great a degree of refinement in the calculation of building heat losses and the use of this information.

Referring to the description of Fig. 3 in this paper, the significance of the seeming increase in efficiency of the larger installations may be noted. It is quite possible that this seeming increase in the case of larger houses could be traced to increasing errors which may affect the accuracy of heat loss estimates of the smaller houses.

The total of 700 installations seems sufficient to warrant placing considerable reliance on the general average data which have been deduced from them. Nevertheless this does not imply that such figures possess absolute accuracy, especially since it is well known to the gas industry that the results of houses in southern territories show lower seasonal efficiencies than those in the north.

DISCUSSION

W. A. OATES[3] (WRITTEN): Of the gas burners installed in our territory they are of three types, the secondary-air controlled, the venturi type of raised port burners and the bunsen tube burners that have no air injection external to the boiler. The latter type combine a form of recuperation of heat from the flue as a part of the gas heating system.

Our conclusions with regard to automatic gas burner operation are substantially in agreement with the facts shown in the paper. The successful operation of these burners rests almost entirely with the care taken in selecting the jobs for installation and the necessity for a careful survey of the heating system to which they are to be connected. This feature applies specially, of course, to the manufactured gas districts with the higher gas rates. There are many installations where it would be inadvisable to connect any automatic heating appliance unless defects in the present heating system were remedied. This applies to both the gas-designed types as well as the automatic burners, but more particularly to the latter because there is the further hazard of the unknown boiler-burner efficiency as against the predetermined efficiency of the gas-designed equipment.

Of equal importance is the installation itself. The controls are similar to those of the gas-designed equipment and produce a low cost for servicing. In all of the gas burner types of today there is but little annual depreciation. The use of refractory baffles to direct the hot gases to the heating surface give an excellent heat application and aid combustion. It is sometimes found desirable to introduce further baffling within the flue-ways but it is usually better to control through the flue collar or properly-sized flue outlet. It was at first thought that in gas fuel there was but little heat from radiation but further research and investigation shows the heat from this source to be much larger than originally supposed. As pointed out in the paper presented, on Orsat analysis or other check is made on every job and the combustion rate set for the proper efficiency. Regular servicing of all gas-fired jobs has a great bearing on successful operation.

The factors used in sizing boilers agree with those of Table 1 but the adjustment of the gas rate has to be determined for the particular job as the response of boiler and heating system together determine this point. Proper combustion can so easily be obtained with gas fuel over wide ranges of throttling that it is possible to adjust the heat input with relation to heating demand more closely than with other fuels.

The results of the survey of 700 residences substantiate those obtained here and contribute reliable data for comparative purposes. I would like to introduce the thought that in any consideration of unit values and particularly those that bear on fuel consumption such consideration would not be complete without taking into

[3] Heating Engineer, Lynn (Mass.) Gas & Electric Co.

account the personal habits of the home owner. In my opinion it is not right to believe they are completely accounted for in general averages. They play a very large part in the seasonal consumption and can outweigh the simple relation between heat input and output. I believe they explain to a large extent the variation in the high and low values. Furthermore, it is my opinion that the theoretical heat loss that must be taken as a unit basis is of itself not a constant relation. Of these variables I feel the principal one is infiltration but we all know that there are many days with cloudy weather and high wind that take much more fuel consumption than those of lower temperature, sunshine and normal wind. With gas metered accurately and the fuel itself of constant Btu value it is possible to obtain consumption figures that give very close results when combined with a flue analysis. When the heat input is compared with the heating demand based on the heat loss and the degree-day figure, day to day comparisons differ widely at certain seasons, principally in the spring and fall. From such results one questions not only the degree day unit, which does not take into account solar and wind conditions, as a means of determination for short period records but also the heat loss values. After testing many installations in this way I have reached the conclusion that unit values are reliable on long period records and may properly show very high efficiency. I have also found, however, that the owners' habits of operation with regard to prevention or non-prevention of waste, temperature operation and partial or full operation of the heating system can be fully as great an effect as the relation between heat input and demand.

N. T. BRANCHE[4] (WRITTEN): There are several points regarding the use of automatic gas burners or conversion burners, as they have been known in the past, which have led to the wide acceptance of this class of equipment during the past several years.

Probably no other factor has contributed to their success more than the fact that they are capable in their modern form of being installed and adjusted to give uniformly good operating results with a minimum of experience necessary. Most of the manufacturers of this class of equipment have simple and straightforward rules to follow in the installation of the equipment and means of balancing and checking drafts, which will give flue analyses and temperatures which are well within the range of that necessary for economic operation.

The second factor which has been of great importance in the picture is that of service and mechanical durability of the equipment as well as that of the automatic control and safety devices incorporated on it. Today, the use of limit controls, safety pilots, gas pressure regulators, thermostats and gas valves is almost certain to be found on every piece of modern conversion equipment, whereas in the past, many of the results tabulated and recorded were taken from inefficient manually controlled jobs without secondary air control and without proper means of utilizing the heat generated within the furnace or boiler proper.

The third point which has also had some bearing on the first one is the matter of operating efficiency with ease of installation. In other words, the absorption of the heat from the flue gases in the equipment which was originally designed for coal and which inherently is lacking in indirect heating surface or secondary heating surface. (For the sake of this discussion, we will call direct heating surface that heating surface which formerly was considered the combustion chamber of the coal furnace or boiler, and the indirect heating surface the passes of sections to which the flue gases must pass before reaching the stack outlet.) The direct heating surface of a coal boiler or furnace is a great proportion of the total, whereas in gas designed boilers and furnaces, it is a very small percentage of the total. In other words, gas designed furnaces and boilers are designed primarily to burn the gas as rapidly as possible in as small a space as possible and then to absorb the heat from the flue

[4] House Heating Engineering Department, Surface Combustion Corp.

gases by use of baffles, passageways and ample heating surfaces before they can escape to the stack.

On coal equipment, we have quite the reverse. Due to the necessity of strong drafts for proper burning of the fuel, the flue passages through the indirect heating surfaces are large, they are not baffled to a great extent and the primary or direct heating surfaces in the fire bowl and combustion chamber are a large percentage of the total. For this reason when gas is burned in a coal designed furnace or boiler, it is necessary to take the greatest advantage possible of the direct heating surface in the fire-pot.

There are two common ways of doing this. The first is the use of a radiant material which is placed in the flame of the burners in such a way that it will dissipate a great amount of radiant energy to the side walls and thus absorb a considerable proportion of the heat input to the burner before the flue gases have escaped from the fire-pot, and the second is by means of turbulence or scrubbing. Probably the most common method in use with oil burners is the turbulent method, whereas the most common use with gas burners has been the radiant method.

Roughly, a surface of one square foot raised to 1600 F will dissipate to a surface at about 200 F, 35,000 Btu per hour. This figure is approximate, but will serve to indicate the amount of heat which can be liberated from a radiant surface without any chance whatsoever of getting this heat back into the flue gases and out the stack, for the reason that the radiant heat will travel through these gases until it strikes a solid without imparting any of its heat to the flue gases. Thus, it becomes apparent that it is not possible to simply install an open flame burner in a fire-pot and expect to get maximum operating efficiency, unless such a burner is capable of setting up turbulent and scrubbing action on the side walls to get intimate contact between all of the flue gases and the heating surface of the furnace or boiler.

The fourth point which has greatly assisted in the acceptance of this class of equipment has been that of the temporary, as it may be called, nature of the installation. The customer feels that he is not wholly committed to gas if he purchases a conversion burner either on a trial plan or a small down payment. He feels that if his gas bills are guaranteed, and they are in many localities, and the guarantee is exceeded, he is free to remove the burner and return it. The gas companies and many of the dealers handling conversion burners have been able to utilize a trial plan because of the fact that the use of automatic heat and the freedom from the labor of attending to the furnace has great appeal to most people and have been able to do this at a relatively small expense per unit installed as compared to straight gas fired equipment.

In comparing the efficiencies of the various types of units which Mr. Segeler has classified, it is to be noted that apparently no classification of warm air equipment has been attempted, and it is noteworthy that through the Middle West and districts which are the larger gas using territories for home heating, warm air furnaces predominate to the extent of 75 per cent to 85 per cent of the total heating plants installed. A survey of the city of Cincinnati showed 83 per cent warm air furnaces, I believe, several years ago.

It has also been found to be a fact that the warm air furnaces converted with a gas burner have proven to give a much higher operating efficiency on a house loss basis than steam boilers so converted. Some idea of the relative volumes of conversion burners, as reported by the gas companies to the *American Gas Association,* may be gained from the following figures. In the year ending July 1, 1930, 176 manufactured gas companies and 31 natural gas companies reported a total of 13,024 gas designed pieces of equipment sold as compared to 20,518 conversion burners. In 1931, 118 manufactured gas companies and 31 natural gas companies reported 7,990 gas designed pieces of equipment as compared to 18,339 conversion burners. It is also well to note that in this 1931 report, of the 18,339 conversion burners sold,

15,153 of them were installed in warm air furnaces, or approximately 82½ per cent, which tallies very closely with the survey mentioned previously made in Cincinnati.

It is also interesting to note in Mr. Segeler's comparison that he states that the majority of the jobs in any one class, as compared to conversion vs. gas designed, were on a par, whereas the 12 per cent difference in average which he has pointed out was largely caused by the installations which fell into the very poor conversion burner class and the very good gas designed equipment. Several years ago, it was a matter of much controversy among the gas industry as to the relative merits of conversion burners, but today the use of them in large quantities on a national basis in both artificial and natural gas fields has more or less stopped this discussion due to the fact that uniformly good results have been obtained.

That there is a logical place in the field for both kinds of equipment has been amply demonstrated by the fact that many more customers have been converted to the use of gas for house heating because of the low initial cost and economical performance of this class of equipment.

W. E. STARK (WRITTEN): The paper that has just been presented is a very comprehensive resumé of the field of application of conversion burners to residential use. Heating homes with conversion burners is very old in the art, and in design and construction such burners have undergone a great deal of improvement since the early days of the application of gas to home heating. The different types of conversion burners available today seem to have about the same operating efficiency regardless of details and complications of construction, and the relation of the operating efficiency of such burners to that of gas-designed boilers and furnaces has been rather soundly established by the experiences of a large number of gas companies at approximately the figure given by Mr. Segeler; namely, 10 per cent added operating cost of the conversion burner as compared with the gas-designed appliance. Individual attention to the conversion burner installation, such as is given by some few gas companies at considerable expense to themselves, or such as has been given in the case of certain widely published tests; may well result in selected conversion installations having operating efficiencies equal to or in excess of the average operating efficiency of the gas-designed appliance. However, complete user satisfaction depends on numerous other things, as well as a few percent gain or loss in operating cost, and the comparison between different types of home heating equipment involves many considerations of which operating cost is but one.

Mr. Segeler well brings out in his paper the impossibility of imposing upon conversion burners the same standards for safety, construction and operation, that are imposed upon and are met by gas-designed appliances. The human element is to be reckoned with in every conversion burner installation, for a given installer is certain to encounter a wide diversity in the characteristics of successive boilers or furnaces in which he is called upon to make installations. There can be no standardization or rigid adherence to well founded principles, such as are followed in the manufacture of a heating appliance including both gas-burning equipment and heat absorbing surface. There is even no absolute certainty that if a conversion burner installation is adjusted for an oxygen content in the flue gases of not less than 4 per cent nor more than 10 per cent, that the products of combustion will be free from carbon monoxide. With some boiler and burner combinations, carbon monoxide can be present in large quantities, even though oxygen in the flue gases falls within the specified range.

Mr. Segeler's remarks about the practice adopted by the *American Gas Association* of allowing a factor of 56 per cent to be added to the installed radiation to take care of starting and piping load. The adequacy of this factor for residential use has been well demonstrated, but it has not been demonstrated that this allowance is unusually large. Therefore, it is hard to see what good reason there can be for applying a lower allowance for piping and starting in the case of conversion burners. If the

factors quoted for conversion burners are correct, then the ones commonly used for gas-designed appliances are too large, a condition which usage has not so far demonstrated.

The gas industry went through a rather well defined history in relation to the cooking load. In the early days of the usage of gas, coal ranges were quite universally converted by installing in them a simple gas burner; and undoubtedly the manufacturer of conversion burners for coal ranges enjoyed great prosperity. Consider, however, the situation today. Can anyone name offhand a manufacturer specializing in the building of conversion burners for coal ranges. or a gas company actively engaged in exploiting such burners? The electric refrigerator industry had something of the same experiences within a shorter period of years. Very few refrigerators are converted today. Any business which specializes in converting an appliance or article designed for the use of a certain source of energy, over to the use of a new source of energy rapidly growing in popularity and acceptance, is hastening its own death; for just as soon as the new source of energy becomes generally accepted, appliances designed solely for its application will assume the ascendency and there will be no more appliances left to convert.

The paper refers to the desirability of a conversion burner causing the least possible change in existing equipment so that the owner will "not feel that he was permanently bound to gas fuel." This is hardly the proper state of mind which a utility should be in when it embarks upon the sale of gas consuming appliances, and is not a thought that the utility salesmen should suggest to the prospects. The utility is interested, primarily, in the sale of energy. It sells appliances only in order that it may sell energy to be consumed in those appliances. It should desire a permanent user of its energy, one convinced from the moment of purchase that he is going to use that particular source of energy forevermore and with no thought in his mind about going back to the old days of coal burning. Experiences seem to indicate that the purchaser of the gas-designed appliance is more likely to consider the use of gas as a permanent policy than is the purchaser of a conversion appliance. Records of removals during this period of depression have substantiated that theory beyond a doubt.

J. M. WILLIAMS (WRITTEN): Quite a number of years ago some investigation was made in the Middle West of installations of conversion burners whereby gas was burned in boilers intended for coal or oil. We found uniform dissatisfaction. It became apparent to us that a boiler to burn gas properly and economically must be designed from the outset to that usage.

More recently we have inquired regarding the use of conversion burners at San Francisco, at which place a large number were installed following the introduction of natural gas some two years ago. We were informed that 90 per cent of the installations of conversion burners had proven unsatisfactory.

The conversion burner appears to offer the most inexpensive method by which gas can be used in heating equipment designed for oil or coal, but it appears that in operation, particularly considering fuel cost, the conversion burner idea has proven far from satisfactory.

H. M. HART (WRITTEN): In reference to the comparative efficiencies of the gas designed equipment and the coal designed equipment the results are just what would be expected for the reason that boilers installed for coal burning are generally selected to give the highest efficiency at the normal or average load while the gas designed boilers are selected to give the highest efficiencies at the maximum load.

When a conversion burner is installed in a boiler that has been installed for coal it must have sufficient capacity to carry the maximum load. When ever there is a demand for heat the burner comes on at its maximum rate of output. This results in low operating efficiency. On the other hand the gas designed boiler being selected

for highest efficiency at maximum load would naturally result in higher operating efficiency.

If a coal designed boiler were selected on the same basis as the gas designed boiler it would probably result in higher operating efficiency than the gas designed boiler. At least the results of the tests conducted at Purdue University would so indicate, because at average loads and intermittent burners the coal designed boilers indicated higher efficiencies than the gas designed boilers.

While we are on this subject I can not refrain from entering a plea to the manufacturers of conversion gas burners to bend their efforts toward the design of continuous rather than intermittent burners for heating boilers and furnaces. The characteristic performance curves for intermittent gas burners is very similar to the intermittent oil burners. The efficiency falls off as the off period is lengthened.

The heat loss from a building is not intermittent but continuous, varying as the difference between the outside and inside effective temperatures vary. Therefore the demand from the heating system is continuous but variable over a very wide range, even greater than the range of outside temperature during the heating season if the building is allowed to cool off during the night or periods of vacancy.

Heating systems are not designed to give uniform heat output with intermittent heat input.

When savings of over 30 per cent can be effected by changing from an intermittent type of oil burner to a continuous type, it seems to me we can not afford to overlook the importance of giving more consideration to the development of continuous burners that are suitable to heating boiler application.

MILTON T. CLOW (WRITTEN): The paper is a valuable contribution to the heating industry in that it clarifies a branch of the industry about which there has been considerable controversy and incomplete information. The timeliness of the information contained in this paper, including data on efficiencies obtained in practice and the conditions governing efficiency, is evidenced by the very rapid growth of the automatic gas burner due to the increased use of lower priced gas fuel made available by extension of natural gas pipe lines. This growth will undoubtedly be much more rapid with a return to normal economic conditions. It may be well to point out that the largest fields for this type of equipment are the low priced natural gas areas, where efficiency is not of great importance. This permits lower installation costs because the installations do not have to have the special treatment required for high efficiencies. Even in manufactured gas territories, there are many people who do not want to scrap their present coal fired equipment which is in good condition, and who feel that their homes will be easier to sell if either gas or coal can be used in the heating plants.

Mr. Segeler has well brought out the inadequacy of reporting comparative efficiencies by the arithmetic average—they should be reported by the mode, that is, the value of most frequent occurrence.

A statement is made in this paper that it has been suggested that a smaller excess capacity can be satisfactorily used with automatic gas burners than with gas designed equipment. Since the reasons for this statement are not self-evident, I would like to hear the explanation.

The matter of estimation of seasonal fuel cost is particularly important to the gas industry, for most people contemplating the purchase of gas heating equipment want a reasonably close idea of what it is going to cost them, particularly since they know it is going to be more than coal fuel. The degree day method of forecasting seasonal fuel requirements has been in use for some years, and my somewhat limited experience with it has shown it to be surprisingly close to actual fuel consumption. However, the data from 700 installations can't be ignored, so that, assuming the

gas input data and the unit heat transmission coefficients to be correct, we must look for possible errors in application of the data and for variable and unrecorded data.

In the first place, it has been generally conceded that the degree day method has failed miserably in the south, and I wonder to what extent the results of the 700 installation survey were influenced by southern installations. Experience has shown that the degree day method under-estimates seasonal fuel consumption in the south, due no doubt, to the following factors:

1. Lower over-all efficiencies caused by more intermittent operation, which in turn is caused by rapidly changing weather.
2. Higher room temperatures than 70 deg are generally required by southern people due to acclimatization to prolonged semi-tropical summers.
3. Greater than calculated infiltration due to inferior construction, and the practice of leaving windows and doors open with the heat on in mild weather.

A further possible explanation of abnormally high gas consumptions, or rather of subnormal heat loss calculations, is the quality of construction workmanship. It is a well known fact the THE GUIDE coefficients are accurate only for what may be considered good workmanship in construction. Obviously, if construction is such that air spaces are subject to ventilation, window cracks are greater than normal and base-board cracks not sealed with plaster, etc., THE GUIDE coefficients will not apply and calculated heat loss will underestimate actual heat loss. Thus, instead of having, say 600 Btu unit gas consumption against 370 Btu required for 100 per cent efficiency, there may be 450 Btu unit gas consumption on the basis of 370 Btu required for 100 per cent efficiency

True, this doesn't account for the exceptionally low gas consumptions, which may possibly be explained by the following factors:

1. *Occupancy:* Is it or is it not a known fact that the occupants of the homes showing subnormal gas consumptions lived in their homes continually throughout the winter? If they were away for an appreciable length of time with the temperature reduced to, say, 60 deg. this factor might explain in some degree at least some of the subnormal gas consumptions.
Particularly during present economic conditions, there is the practice of closing off part of the house and only heating the most frequently used rooms.
2. *Number of Hours at 70 deg.:* The unit fuel requirements in this paper were based on maintaining a temperature of 70 deg. for 24 hours a day, but usually, temperatures are reduced at night. The unit gas consumption of 370 Btu per degree day per 1000 Btu hourly loss would be changed to 350 by maintaining 70 deg. for 16 hours and 60 deg. for 8 hours.

The degree day method assumes that certain variable and unrecorded data will cancel themselves over a season's operation, but let us see if this is true.

The heat lag factor due to the heat capacity of walls probably would cancel itself—sudden changes from a low to a high outdoor temperature offsetting the benefits derived by a sudden change from a high to a low outdoor temperature.

However, the sun effect factor is not considered in transmission coefficients and when it occurs, represents a saving in fuel required. Incidentally, this factor is most pronounced with poor construction.

Unit heat transmission coefficients are based on a 15 mile per hour wind velocity, and although this factor may cancel itself in localities where the wind velocity averages 15 miles per hour, there is no cancellation in localities where it averages much lower. In such localities actual fuel requirements are bound to be less than requirements calculated from coefficients based on 15 miles per hour wind velocity.

In conclusion, a very important question is, "What is the basis for the estimate of fuel consumption?" Is it the calculated heat loss of the building, or is it the capacity of the heating units installed, as determined by heat loss calculations? That is, when the heat loss of a building is calculated to determine the size of heating plant to install, certain percentages are added to normal heat loss to allow for greater than normal wind velocity, and also as a factor of safety for rapid heating, etc. Naturally, the normal heat loss and not the capacity of the heating plant should be used for estimating fuel requirements, and if the reverse practice is used, seasonal

fuel requirements will be overestimated. Thus, this factor could account for the subnormal gas consumptions shown in this paper.

In summary, abnormal gas consumptions may be explained by the following factors:

1. Influence of southern installations.
2. Quality of construction.

Subnormal gas consumptions may be explained by the following factors:

1. Occupancy.
2. Partial heating of building.
3. Number of hours temperature is maintained at 70 deg.
4. Sun effect.
5. Actual average wind velocities lower than 15 mi. per hr.
6. Basis of fuel consumption estimate.

Some of these factors, if they do exist in the data from the 700 installations, represent avoidable errors which may reduce variations to a reasonable figure. If anything, the factors indicate the need for still further refinement of methods for calculating and estimating heat losses and heat requirements. True, this further refinement means more time required for selling a heating plant, but first let us show that accuracy can be attained. Then, whether or not gas engineers will be willing to use the available data and spend more time calculating and estimating will be determined by the *need* for accuracy.

MR. SEGELER: I have carefully noted Mr. Clow's report and would like to add that the survey mentioned in my paper was not measurably influenced by extreme Southern installations. There were 40 installations from Memphis and 100 from Washington, D. C.; all the rest of the data was from territory well North of the Mason-Dixon Line.

E. D. MILENER: I want to point out that every gas-heating installation is essentially a field laboratory because the fuel is delivered automatically and is as accurately measured as it is in the laboratory. When the data resulting from these extensive field laboratory tests vary as much as they have in this paper, it certainly presents a problem to us which causes us to reach the same conclusion that Professor Willard reached, namely that laboratory conditions don't always hold but that field conditions must be studied and conclusions based more on operating conditions.

MR. JOHNS: The paper calls attention to the need for watching the oxygen content of flue gases and I should like to emphasize that point by saying that it is just as important in automatic gas burner installations as it is with any other fuel. The percentage of carbon dioxide in the flue gases as compared to the theoretical must be watched.

A point which might lead us to the wrong inference is that a very low flue temperature may be expected with a low oxygen content in the flue gases. Our experience has indicated on many occasions that as we reduce the oxygen content in our flue gases we actually increase the flue temperature, but we are improving the operating efficiency in spite of the higher flue temperature. In other words, the heat carried up by the excess air is much less due to the decrease in oxygen content (and of course nitrogen) in spite of a higher flue temperature.

The author calculates that the theoretical amount of fuel required to heat the house at 100 per cent efficiency would be 370 Btu per degree day per thousand Btu hourly heat loss from the house. The 100,000 Btu example is figured from zero to 70 deg and then it is explained that a zero day will have 65 deg days.

I think first of all we should define a degree day. Perhaps that has been done, I don't know, but my understanding of a degree day is that it is a day in which the average outside temperature is 1 deg less than the average inside temperature. Therefore, if there are 65 deg days on a zero day, the average inside temperature is not 70 deg as the calculation is based on but is 65 deg and the heat loss would

be not 24 times 100,000 but 24 times 65 seventieths of a hundred thousand. That would be my understanding and on that basis his figure of 370 should become 343.

Bearing this in mind when studying the cases which indicate 140 to 150 per cent seasonal operating efficiency, we cut the 140 down to about 125 and the 150 down to about 135 per cent.

We still have the problem that Mr. Segeler presented of reconciling those efficiencies that are over 100 per cent, but I think this would help to reconcile it, if I haven't gone astray in my figuring.

E. A. JONES: I have a thought relative to Fig. 5. On many hundred checked installations we have found very good agreement between the calculated fuel consumption and operating results. On the other hand, of course there is the variation as indicated on this chart. However, I am inclined to believe that without giving due consideration to the general agreement, the chart is apt to be misleading. Actual fuel consumptions in the main agree very well with calculated fuel consumptions, and I would suggest that this chart be revised by plotting the dispersion into blocks very much as we plot an artillery dispersion scale. In other words, if they would put down a dot at each efficiency for each job in these little blocks here, they would then secure a shading which would indicate the average.

I am inclined to think that the indicated average in the case of the gas-designed appliances would be fairly well towards the lower end of the block, and I am inclined to believe that it would also indicate that we could place more confidence in our calculations than might be indicated by simply studying Fig. 5. That is simply a suggestion for an amplification.

MR. CONNER: To the gas industry this is a very important subject. I can't say however that it might be of quite as great interest to this Society. I think that there have been more misstatements made about conversion burners than any other type of gas heating appliance. Conversion burners have a legitimate field, a very important one, and I don't think there is any one in the gas industry that would want to do or say anything that might interfere with the extension of that class of heating. On the other hand, I think it is easier to go wrong on a conversion burner installation than any other kind of a gas appliance heating job. There are so many variables that must be taken into consideration that practically every installation as has been pointed out by Mr. Milener, is a separate engineering job.

Many tests made by various institutions and private laboratories on conversion burners in the main has given information that has been misleading. It generally refers to a conversion burner installation that was made under more or less ideal conditions with drafts very carefully controlled, with the secondary air, excess air, etc., properly regulated and an effective chimney, properly dampered and a burner well adjusted without any variations in gas pressure or heating value.

Consequently the information that has been brought out, seems to me to have put the conversion burner in a more favorable light than it is really entitled to. This is a rather definite statement and I want to go back and say again that we in the gas industry want to do everything that we can to promote legitimate heating business, but at the same time we don't want any one particularly in the heating profession to be under any misapprehensions. Every job must be very carefully engineered if we are to get the proper kind of service. You can't just go out and stick a gas burner in some old antiquated boiler or furnace and expect to get a 100 per cent installation.

MR. MILENER: I am very glad that Mr. Conner brought out his point so forcefully. Undoubtedly a further check-up on the data gathered will disclose that Mr. Jones is correct, and I believe that the data should be revised to take that into consideration.

The interesting thing though, in that respect, is that of the large number of individual installations reported all the buildings were calculated on a standard method of heat loss. We know the gas was accurately measured, but yet a number of calculated efficiencies went beyond 100 per cent. I have wondered whether or not the differences in the amount of humidification in the individual houses wasn't largely the cause.

With regard to Mr. Jones' suggestion that these data, particularly Fig. 5, should be shaded according to the number of installations, I believe that the original data were fairly constant. I can recall in 1928 that I reported in a paper on forty installations, ten gas-designed for hot water, ten gas-designed for steam, ten converted hot water and ten converted steam. They were all installations that had been installed 10 years before and the conclusions drawn in that paper at that time were somewhat the same as the conclusions drawn in this paper, only they applied to a limited number of installations.

This paper is not an attempt in any sense to point out the relative merits between automatic gas burners and gas-designed equipment. I don't think that any of us are in a position at this time to definitely point out in a concise statement the relative merits of the two. They both have their fields. The field for gas-designed equipment is a large one and it is a growing one because there is no doubt that in the long run the gas-designed equipment does have many advantages which you cannot incorporate in an individual burner.

Mr. Conner's remarks about the discretion that an installer has to use is probably a familiar one to all of you and it is only through the same discretion that proper and satisfactory installations will result.

In Memoriam

Names	Joined the Society	Died
Harry W. Arthur	1920	Apr. 1932
Robert B. Beahm	1919	Aug. 1932
J. Esten Bolling	1918	June 1932
Harry F. Bray	1932	Sept. 1932
James J. Brogan	1917	Sept. 1932
W. B. Crawford	1921	Feb. 1932
James H. Davis	Charter Member	Jan. 1932
John A. Flemings	1929	July 1932
Benjamin E. Haskell	1925	Dec. 1932
Otto Helphingstein	1919	Sept. 1932
George D. Hoffman	1906	Sept. 1932
Charles R. Honiball	1911	Apr. 1932
Eubertis L. Jaynes	1918	Aug. 1932
August Kehm	1901	Dec. 1932
George H. Kirk	1906	Mar. 1932
J. Stuart Knee	1931	Sept. 1932
George Latham	1924	June 1932
George Mehring	Charter Member	June 1932
T. H. Monaghan	1914	Aug. 1932
Raymond Newcomb	1924	Feb. 1932
George P. Smith	1922	Aug. 1932
Louis J. Sommer, Jr.	1922	June 1932
Paul E. Sowers	1922	Oct. 1932
James F. Sprague	1930	Dec. 1932
Charles J. Steim, Jr.	1923	July 1932
William W. Underhill	1913	June 1932

INDEX

A

B

D

E

F

I

J

K

L

M

N

O

V

W

Y